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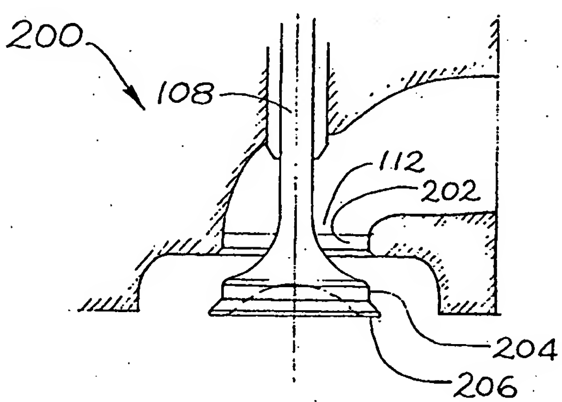

(57) Abstract

An improved poppet valve (200) provides quick opening characteristic that allows for easy breathing while avoiding the refrigerating loss of heat associated with conventional poppet valves. The improved poppet valve includes a valve stem (108), a valve head (206) coupled to the valve stem wherein the valve head further includes a substantially cylindrical portion (204). This substantially cylindrical portion (204) slideably mates with a corresponding substantially cylindrical mating surface (202) that defines a part of the aperture of the port (112) in a housing — typically, an inlet or exhaust port of an internal combustion engine. The substantially cylindrical portion (204) makes a seal with the substantially cylindrical mating surface (202) of the port such that gases that flow through the port do not see an effective opening to either enter or exit the cylinder until the substantially cylindrical portion (204) has cleared the substantially cylindrical mating surface (202) of the port. In another aspect of the present invention, the improved poppet valve employs an adjustable valve timing shedule and valve actuation that compensates for dynamic load differentials as well as providing for altitude compensation.

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VALVE SYSTEM HAVING IMPROVED OPENING AND BREATHING CHARARCTERISTICS FOR INTERNAL COMBUSTION ENGINES

Technical Field

The present invention relates to valves for the intake and exhaustion of gases through ports and, in particular, to fast opening poppet valves having improved breathing characteristics.

Background Art

One problem with the conventional poppet valve 100, as shown in **Figure 1**, is that the current design creates a "refrigerating" effect -- as the slow opening characteristic of the poppet requires that the gases squeeze through a narrow opening in the beginning portion of its opening cycle. As seen in **Figure 1**, cam 102 rotates clockwise, pushing against cam follower 104 to compress spring 106, thus inducing valve stem 108 to move downward. This movement causes poppet head 110 to move off of valve seat portion of opening port 112 of the cylinder -- thereby allowing gases to exit.

With the cam revolving typically as fast as fifty times a second, very high forces are created to overcome the inertia of the valve. The closing spring must be strong enough to close the valve when the cam passes and the cam must provide enough force to open the valve against its own inertia plus force the spring down. It is evident that the profile of the cam is very critical so that it will lift the valve off its seat gradually when opening and lower it gradually to the seat when closing to minimize seat hammering and valve bounce.

The gradual opening and closing of the valve is essential to ensure that the mechanism will survive at high speed, but it incurs a slow opening and closing quality to the valve operation. In order that the valve will breathe easily at high speed, its open duration is made considerably longer than would be ideal for good low speed operation. This compromise has been necessary in poppet valve engines

because of an essential weakness of the valve -- that it is stationary at the instant it begins to open and finishes closing.

As mentioned in the previous patent application, the piston valve, by contrast, is in rapid motion the instant it opens and closes -- by the action of its linkage. Thus, its rate of opening and closing is far greater than that of the poppet valve without incurring high mechanical forces. The piston valve can utilize the short open duration while allowing freer breathing at the high speed end of the engine operating spectrum. Thus, while the piston valve is open only approximately 83% as long as a conventional poppet valve, its breathing ability is approximately 150% of the conventional poppet valve.

As can also be seen from **Figure 1**, the geometry of poppet head 110 gives a clue as to the reason for the aforementioned "refrigerating" effect that reduces the amount of heat of the exiting gases. Mating surface 114 of poppet head 110 is typically at a 45 degree angle; thus gases can begin to escape upon the first downward movement of valve stem 108. The opening afforded by the poppet valve is small initially, however, as compared with the opening afforded by the poppet when fully opened (i.e. when cam 102 has rotated so that furthest point from its center is at "6 o'clock"). Thus, at the first part of its opening cycle, the aforementioned opening is very small and the entering or escaping gases "squeeze" around that small opening, thereby inducing some refrigerating loss of heat in the entering or exiting gases.

This slow opening produces a throttling loss, reducing the overall efficiency of a conventional internal combustion engine. This loss, moreover, is most acutely felt in a compound engine design where the heat content of the exiting gases from the primary cylinder is required to perform positive mechanical work against the piston of the secondary, expansion cylinder. This leads to a less efficient engine.

Thus, there is a need for a valve system that has substantially quick opening characteristics; but without the need for redesigning a portion of the head which has become standard and ubiquitous in today's conventional engine.

Apart from the loss of useable heat content of the exhaust gases at the exhaust port that might be utilized by a compound engine design, there is another source of inefficiency that occurs in even conventional engines at the inlet port -- i.e. pumping

losses. Pumping loss (and throttling loss, in general) in the engine may be defined as work subtracted from the engine's output required to drive gases through a restricted opening. The quantity of work subtracted from the engine's output is proportional to the mass of gasses (in weight per unit time or pounds per minute) multiplied by the pressure drop across the restriction (i.e. pressure upstream of the restriction minus pressure downstream from it).

When the engine is operating at wide open throttle and at intermediate rotational speed, these losses are small, possibly two or three percent of the engine's output. However, under most driving conditions, the engine is developing perhaps not more than ten to fifteen percent of its maximum power capability. Under these conditions, the pumping losses do not diminish proportionally and may consume a great deal more than two to three percent of the small amount of power being produced at that time.

The largest pumping loss during valve opening comes when the exhaust valve opens because, at full power, the cylinder pressure is of the order of 80 psi above atmosphere while the pressure downstream is only on the order of 1 to 2 psi above atmosphere. At light load, the cylinder pressure may drop to about 20 psi above atmosphere at the moment the exhaust valve opens, which is proportionally greater than the amount the engine's power is diminished.

By improving breathing at the inlet valve, improvements may be made in the efficiency of the engine by providing a broader power range which includes a slow, smooth idle, good pullup torque and good intermediate speed pulling power. Conventional engines sacrifice some of these desirable characteristics for the sake of very high maximum power.

For example, to achieve very high horsepower, one requirement is to fill the cylinder as full as possible with fuel vapor during the inlet stroke. One long used method is to supercharge or pump in fuel vapor at above atmospheric pressure. This method, though effective, is not often used on passenger cars because of increased cost and complexity. A more universal means of getting more fuel vapor into the cylinder at high speed is to leave the inlet valve open well past the time the piston has reached the bottom center. This is effective because during the inlet stroke,

especially at high engine speed, the charge of inlet air or fuel vapor in the inlet manifold achieves considerable velocity and therefore gains inertia. If the inlet valve remains open, the air or vapor will continue to hurl itself into the cylinder even as the piston starts up on the compression stroke.

To allow this to be effective at high speed, the inlet valve is, in some engines, held open until 65 degrees past bottom center. On the Honda VTEC engine in the high speed inlet cam mode, it is held open even longer. This typical late closing of the inlet valve aids in producing high power at high RPM, but it is detrimental to the engine's smooth, efficient low speed performance because at low speeds, the incoming charge of air or fuel vapor has very little velocity, and will not stuff itself into the cylinder much after the piston reaches bottom center.

Instead, if the inlet valve remains open as the piston begins the compression stroke, some of the charge will be pushed back into the inlet manifold. Thus, under low and moderate engine speeds, the pistons are compressing less than a full charge of vapor and the engine will feel listless and more feeble than it would if the inlet valves closed shortly after bottom center. Hence, under these conditions, the Honda VTEC engine switches to different inlet cams which close the inlet valves earlier in the compression stroke.

Thus, there is a need for a valve system that opens quickly, allowing the cylinder to fill and breath easily during the inlet stroke and yet closes shortly (about 20 degrees) after bottom center, thus preventing spitback (i.e. loss of some of the charge) during the compression stroke.

There is also a need for a valve system that improves efficiency and lower emissions in either a conventional internal combustion engine or a compound engine.

There is also a need for a valve system that allows the engine to go to higher speeds since closing time can be extended by the cam design, resulting in higher power for a given engine displacement.

It is known for some engines to be designed so that the compression ratio differs from the expansion ratio to increase the overall thermodynamic efficiency of the engine. Two styles of engine design – the Atkinson cycle engine and the Miller cycle engine – are the most notable of such design. An Atkinson cycle engine is one

in which the actual stroke length of the expansion stroke physically differs from the compression stroke of the engine. Such difference in stroke length is effective by a linkage that physically alters the stroke length, as is well known in the art.

The drawbacks to the Aktinson cycle engine are also well known. They require additional linkage that is subject to breakdown and wear. They also increase the cost of the engine due to the added number of parts.

A Miller cycle engine attempts to achieve the same difference in "effective" stroke length of the compression and expansion stroke by altering the timing of the inlet valve – without use of additional linkage. It is known in conventional Miller cycle engines to either close the inlet valve "before" Bottom Dead Center (bBDC) on the inlet stroke or to leave it open "after" Bottom Dead Center (aBDC) on the inlet stroke.

In a conventional Miller cycle engine, the inlet valve is a conventional poppet valve or valves. As discussed in the parent application (Serial No. 09/307,963) which is hereby incorporated by reference, the convention poppet system has some inherent drawbacks in its breathing characteristics. These drawbacks are also apparent in the operation of a Miller cycle engine with conventional poppet inlet valves.

Thus, there is a need for a valve system that effects a Miller cycle operation that is efficient from a breathing standpoint.

Additionally, there is a need for a valve system to provide a more nearly continuous range of inlet opening duration and amount to provide flexibility over a wide range of driving conditions. Such driving conditions include flexible power demands by the driver and altitude compensation without need for throttling or other restrictions.

Disclosure of Invention

The present invention meets the aforementioned needs by providing an improved poppet valve that provides a quick opening characteristic that allows for easy breathing while avoiding the refrigerating loss of heat associated with conventional poppet valves. The presently claimed poppet valve comprises a valve

stem, a valve head coupled to said valve stem wherein said valve head further comprises a substantially cylindrical portion. This substantially cylindrical portion slideably mates with a corresponding substantially cylindrical mating surface that defines a part of the aperture of the port in a housing -- typically, an inlet or exhaust port of an internal combustion engine.

The substantially cylindrical portion makes a seal with the substantially cylindrical mating surface of the port such that gases that flow through the port do not see an effective opening to either enter or exit the cylinder until the substantially cylindrical portion has cleared the substantially cylindrical mating surface of the port. At that point, the poppet valve is traveling at an initial, non-zero velocity; and, the gases experience a very quick opening through which to either enter or exit. Thus, the refrigerating effect incurred by conventional poppet valves is obviated.

In another aspect of the present invention, the improved poppet valve comprises both a substantially cylindrical portion and a conical portion. The port aperture has a corresponding substantially cylindrical mating surface and a conical mating surface so that the poppet valve mates with both portion when closed. The conical portion may be placed above, below, or in between substantially cylindrical portions.

In yet another aspect of the present invention, improved poppet valve may further comprise a relief groove and/or a gas trapping groove for improved manufacturing and performance.

One advantage of the present invention is performance. The improved breathing characteristics of the present invention reduces both the refrigerating losses as well as pumping losses associated with conventional poppet valves.'

Another advantage of the present invention is ease of manufacturing. The improved poppet valves of the present invention give internal combustion engines substantially the same improvement in breathing characteristics and performance as afforded by piston valves. However, there is no need to redesign the cylinder head to accommodate the improved poppet valves as would be required for piston valves. Additionally, the other components of the poppet valve system, including cam system, is essentially the same as the convention poppet valve system.

In addition to the above, the present invention meets the aforementioned needs by providing an improved poppet valve that provides an improved Miller cycle operation for an internal combustion engine. The presently claimed Miller cycle operation employs the aforementioned fast opening poppet valves, together with an adjustable valve timing schedule and valve actuation that compensates for dynamic load differentials as well as providing for altitude compensation.

In one embodiment of the present invention, the adjustable valve timing and actuation is implemented via a unique linkage that dynamically changes the duration of valve opening as well as valve lift, as driving conditions dictate. In another embodiment of the present invention, the adjustable valve timing and actuation is implemented via a solenoid, electromagnetic, pneumatic or hydraulic system.

One advantage of the present invention is performance. The valve system of the present invention dynamically adjusts to changes in the prevailing driving conditions. Such conditions might be dictated by power demands by the driver and altitude compensation, among other factors. Thus, the power output might be as high at 12,000 feet as it is at sea level, possibly without need of supercharging in the inlet charge.

Another advantage of the present invention is efficiency. The employment of the fast opening poppet valve herein disclosed enables the valve open duration to be shorter than the cam duration. Additionally, the mechanism of the present invention allows the open duration to be varied from near zero (e.g. idling) up to the full duration of the cam for full power.

Other features and advantages are disclosed in the specification below and in conjunction with the accompanying figures.

Brief Description of Drawings

Figure 1 is a prior art view of a conventional poppet valve.

Figure 2 shows one embodiment of the poppet valve system as made in accordance with the principles of the present invention.

Figures 3 A, B, C show three embodiments of the presently claimed poppet valve in which the conical portion is below, above, and in between substantially cylindrical portions, respectively.

Figure 4 shows one embodiment of the presently claimed poppet valves employed inside an internal combustion engine.

Figure 5 shows one embodiment of the presently claimed poppet valves employed inside a compound internal combustion engine.

Figures 6A-6H show various embodiments of the presently claimed poppet valves.

Figure 7 shows a close-up view of the presently claimed poppet valves slideably coupled to conical and substantially cylindrical mating surfaces of the port opening.

Figure 8 is a close-up cross-section view of the presently claimed poppet valve having rounded corners coupled to the port opening.

Figures 9 A, B, C show three different embodiments of the present invention wherein the axial length of the substantially cylindrical portion need not be the same axial length of the substantially cylindrical mating surface.

Figure 10 depicts two opening performance curves, one of a conventional poppet valve and the other of the presently claimed poppet valve.

Figures 11A-11D depict various inlet valve timing diagrams to effect a Miller cycle operation.

Figure 12 is a graph of the opening characteristics of the fast opening poppet inlet valves effecting a Miller cycle operation under various driving conditions.

Figure 13 shows a novel linkage that effects the Miller cycle operation of the present invention.

Figure 14 depicts the operation of the novel linkage as shown in **Figure 13** to effect a dynamic response of the Miller cycle operation under different driving conditions.

Figure 15 depicts the fast opening poppet inlet valve as operated by a solenoid or hydraulic controls.

Figure 16 depicts a control mechanism built in accordance with the principles of the present invention.

Best Mode for Carrying Out the Invention

Referring now to **Figure 2**, a side view of one embodiment of the present invention is shown. Poppet valve 200 comprises valve stem 108, mating surface 204, and poppet valve head 206. As seen in **Figure 2**, mating surface 204 comprises a substantially cylindrical shape (e.g. comprising sides that are substantially parallel with the axis of valve stem 108). Mating surface 204 substantially matches a corresponding surface 202 on the opening port 112. In a preferred embodiment, the clearance between the mating surface 204 and corresponding surface 202 is small, so that leakage of gases is preferably small due to the relatively low pressures involved during the short time between the separation of the mating surface 204 and corresponding surface 202.

In practice, however, it may be necessary to design the cylindrical portion of the claimed poppet valve so that, at cold start, the cylindrical portion has greater clearance with the cylindrical mating surface; and thus, does not make a perfect seal. As the temperature of the poppet valve achieves temperature equilibrium in an operating engine, both the poppet valve and the opening port will expand according to their respective thermal coefficients. It is thus preferable to size the poppet and the opening port and to select their materials appropriately, so that at thermal equilibrium, the seal is as good as possible. The materials of the valve and port may be of the same material as those presently used since they are proven to be compatible in the conical seating area. Also, these materials have a low thermal coefficient of expansion. The radial clearance afforded at full operating temperature may be the same as the valve stem to guide clearance so that no rubbing will occur. This radial clearance may be of the order of 0.002 inch per inch of valve head diameter. In any event, the initial clearance should take into account the worst case conditions, where the engine runs at high power from cold startup. In such a case, it is likely that the valve will enlarge before the seat does. The cold clearance leakage,

however, should not be problematic after warmup; but during warmup, the benefits of the valve may be slightly affected.

It will be appreciated that other materials and dimensions might be suitable for the purposes of the present invention. for example. materials having lower thermal coefficient of expansion (e.g. ceramics) could be used to reduce any leakage that might occur because of the initial radial clearance to avoid rubbing. A lower thermal coefficient of expansion of the material might allow a smaller radial clearance. thereby reducing initial leakage. As the materials heat up to operating temperatures. the materials will expand -- decreasing the radial clearance. and thus. decreasing the initial leakage.

So designed. the depth of mating surface 204 is generally -- but not necessarily -- the initial distance that the valve stem moves before gases are allowed to view an opening. allowing gases to either enter or exit the cylinder. As the valve is already in motion (and more generally. accelerating) by the time this opening is viewed. the gases experience a quickly opening hole. As with the piston valve. this quickly opening aperture alleviates the refrigerating effect and energy loss of hot gases squeezing through a small aperture; thereby preserving the amount of useable heat in the exiting or entering gases that would have otherwise been lost.

Although different depths are suitable for the purposes of the present invention. a typical depth of about 0.080 inches (or about 2mm) will suffice in a typical application where the cam rotates fifty times a second with a displacement of 0.3 to 0.5 inches (8 to 12 mm). One suitable depth range might be approximately 5 percent to 40 percent of the valve lift (with 15 to 25 percent as possibly preferable for purposes of the present invention); thus. the cylindrical portion acts like a short piston valve and allows the cam time to get the valve moving with considerable velocity the instant the cylindrical portion clears the cylindrical bore of the opening port. The improved poppet valve thus combines the good sealing against great heat and pressure of the conventional poppet valve with the very rapid opening and closing properties of the piston valve.

It will be appreciated that it merely suffices for the purposes of the present invention that some distance be traveled by the poppet valve -- thereby achieving

some velocity -- prior to the gases experiencing an opening; and as such, the scope of the present invention should not be limited by any particular depth. For example, variation in the depth of the substantially cylindrical section might be indicated by experience and would probably be less on low speed engines. Thus, depth of the cylindrical portion might, in some circumstances, be lower than 5 percent of valve lift, or, alternatively, greater than 40 percent of valve lift. In fact, the cylindrical portion on the valve head might even be a sliver; it suffices for purposes of the present invention that good sealing be maintained between the cylindrical portion on the valve head and the cylindrical mating surface of the port. Thus, gases do not experience an effective opening until the valve has achieved an initial velocity for quick opening.

Valve head 206 is a substantially conical sealing portion that provides the final sealing and closing of the valve. It will be appreciated that such a conical portion may precede or follow the cylindrical portion, be between two cylindrical portions, or may be eliminated altogether. It suffices for the purposes of the present invention only that there be a geometry of the mating surface of the poppet valve that provides a reasonable seal against gases during an initial portion of the travel of the poppet valve. Additionally, the poppet valve itself might be hollowed out to reduce the weight and inertia of the valve; thus, reducing the force necessary to provide a given acceleration.

Other embodiments of the present invention are shown in **Figures 3 A, B, C** whereby the conical mating surface is either below, above, or between substantial cylindrical sections. As before, poppet valve 200 comprises valve stem 108, valve head 206, and substantially cylindrical mating surface 204. Opening port 112 comprises a corresponding mating surface 202 comprising either a conical and substantially cylindrical surface, as needed. Here, as discussed above, the conical mating surface of valve head 206 may be below, above, or between substantially cylindrical portion 204 of the poppet valve as shown in **Figures 3 A, B, C** respectively.

The depth of the substantially cylindrical mating surface 204 is the initial distance poppet valve stem 108 must travel before gases may view an opening by

which they may either enter or exit. As before, as poppet valve 200 thus has an initial velocity (and a positive acceleration), the opening characteristic of poppet valve 200 is such that it is quicker than a conventional poppet valve. Having a conic mating surface -- either above, in between, or below -- a substantially cylindrical portion may be advantageous in helping to form a final seal and to help withstand the very high pressures experienced by the poppet valve during combustion -- perhaps as high as 1000 psi in a gasoline engine or 1600 psi in a Diesel engine. However, for purposes of the present invention, the conic mating surface might be eliminated altogether from the valve head. A suitable conic mating surface or any other stop might then be placed elsewhere on the valve (e.g. at the top of the valve stem) to prevent the combustion pressure from damaging the cam mechanism.

Figure 4 shows the present invention functioning in a conventional internal combustion cylinder. As noted, the leftmost poppet valve 108 is opening to allow outside air to enter the cylinder, as in a typical intake stroke. As previously discussed, the quick opening characteristics of this poppet valve allows fresh air to enter more quickly than a conventional poppet valve. As noted above, one advantage of using the presently claimed poppet valve on the inlet port of a cylinder is performance, enabling the engine to perform strongly and efficiently over a broader speed range which includes a slow smooth idle, good pullup torque and good intermediate speed pulling power. Modern conventional engines sacrifice some of these desirable characteristics for the sake of very high maximum power.

Figure 5 shows the present invention functioning in a compound engine, such as discussed in the aforementioned incorporated patent application. As shown, primary cylinder 510 is under going an exhaust cycle and intermediate poppet valve 204 is opening to allow hot exhaust gas to exit cylinder 510, so that it may traverse intermediate catalytic converter 514 on its way to secondary expansion cylinder 512 to do further mechanical work. As reported in the incorporated patent application, the intermediate catalytic converter 514 is used to clean the air by completing the combustion of the exhaust gases, and thereby increase the heat content of the gases entering the secondary cylinder.

As previously discussed, the quick opening characteristic of poppet valve 200 avoids the refrigerating heat losses that the conventional poppet valve would induce. This improves the efficiency of the compound engine by retaining valuable heat that may be converted into useful work in the secondary cylinder.

The compound engine of **Figure 5** actually comprises two primary cylinders (second primary cylinder is behind the first primary cylinder, and is not shown) connected to catalytic converter 514 via a common manifold. The two primary cylinders exhaust alternatively into the secondary cylinder in a typical four stroke-two stroke configuration. Thus, the primary cylinder shown in **Figure 5** is starting its exhaust stroke while the second primary cylinder is starting its compression stroke.

Since both exhaust ports of the primary cylinders are closely connected where they enter the catalytic converter, a puff of high pressure from one primary cylinder exhaust valve may blow open the exhaust valve of the other primary cylinder and ignite the inlet charge within that cylinder.

However, the fast opening poppet valve herein described prevents this occurrence because if the puff of high pressure gas is applied in the valve port, it may be sufficient to force the valve off its seat against the closing force of the spring, but in order for the valve to "open", the gas pressure behind it must force the valve farther open until the cylindrical portion clears the cylindrical bore. However, as the valve is forced down, the force of spring action increases proportionally to the distance displaced. Thus, it is merely sufficient that the spring be selected so that the back pressure of the exhaust gas does not force the other poppet valve of the second high pressure cylinder to be "open".

Figures 6A - 6H depict several alternative embodiments of the present invention. **Figure 6A** shows a poppet valve comprising two substantially cylindrical portions 602 and an intermediate conical portion 604. This design has the advantage of providing better sealing without increasing the lift of the valve. **Figure 6B** depicts a similar valve to the previous one; but where the substantially cylindrical portions 602 are actually cones that are displaced at angles off the axis of the valve stem. It will be appreciated that these angles may be different and that the individual angles themselves may exhibit an effective range whereby there is a reasonable seal for

gases either entering or exiting for an effective range of displacement of the valve itself. This reinforces the desirable feature of the present invention that the poppet valve have a quick opening characteristic -- accomplished by allowing the poppet valve to have an initial non-zero velocity before the gases see an appreciable opening through which to escape. Such a range of angles might vary depending upon a number of factors, including the depth of the substantially cylindrical portion, the desired velocity of the poppet valve before gas escape, among other factors. Preferably, this angle is substantially zero (i.e. approximating a truly cylindrical portion), but adequate performance might be had in the range of zero degrees to 30 degrees, assuming the depths are 4 to 40 percent of valve lift.

Figure 6C shows yet another embodiment of the present invention wherein a relief groove 606 is provided between two portions of the valve -- in this case, the two portions are conical portion 604 and substantially cylindrical portion 602. This relief groove would facilitate manufacturing by allowing clearance for the grinding wheel corner.

Figure 6D shows another embodiment of the present invention wherein a gas trapping groove 608 is intermediate to two substantially cylindrical portions 602. This type of groove reduces the gas leakage past the face by creating turbulence in the leakage gases. Also noted, conical portion 604 exhibits a cone angle that might vary from 20 degrees to 90 degrees to provide adequate sealing. The closer the angle is to 90 degrees, the valve effectively seals on a flat face, which may be desirable in a few applications. A typical valve is 45 degrees as this has demonstrated good results in most engines.

Figure 6E shows the configuration as depicted in **Figure 6A**; but with two gas trapping grooves 608 cut, or otherwise formed, into the two substantially cylindrical portions. **Figure 6F** shows gas trapping groove 608 cut (or intermediate) into substantially cylindrical portions 602 in the form of a helix or some other pattern (e.g. thread-like). It will be appreciated that the gas trapping grooves are not necessarily confined to a single groove, but may be multiple and may be round bottomed, square bottomed, conical or helical. They may also be made in the bore rather than the valve itself, or in both. **Figure 6G** shows essentially the same

configuration as **Figure 6E**; but with the gas trapping grooves cut, or otherwise formed, as two helical grooves.

In any of the valves shown above, corner radii or "fairing" (element 610, as shown in **Figure 6H**) may be employed to streamline the gas flow and prevent hot edges which sometimes lead to pre-ignition. Other features of any of the aforementioned embodiments might include: a front area (such as 208 in **Figure 2**) which may be hollowed out; thereby reducing the weight of the poppet valve and hence reducing the force necessary to achieve a given acceleration -- to improve its quick opening characteristic. Additionally, the valve itself may be hollow and cooled with sodium or some other well-known medium. This would improve heat transfer from head to stem where heat disperses. In practice, about half of the heat disperses through the cooled conical seat and about half through the stem to the cooled guide. Alternatively, a copper or aluminum core pin may be incorporated into a valve to increase heat conductivity from head to stem.

Other features that have evolved to improve conventional poppet valves may also find usefulness in the present invention. For example, Stellite or other hard seat materials, chrome plated or molybdenum disulphide coated stems, hard stem tip buttons, helical stem lubrication grooves, among others, may be applied to the present invention.

Figure 7 shows a closer view of the presently claimed poppet valve seated in its port. Conical seating portion 604 is depicted as having a certain width W. As the valve is held against the seat by the valve spring, heat is transferred from the valve to the seat at this time. If the width of the seat is too wide, then the unit pressure of the hot valve head against the seat may be insufficient for good heat transfer. Likewise, if the seat is too narrow, it may not wear well and again the heat transfer may be less than ideal because the area of contact is too small. In practice, the valve seat width may assume a working range of widths; but preferably should be four to six percent of the valve head diameter.

Figure 8 shows a blown-up cross section of the valve in its seat. Corners 810 on the conical seat portion 610 are shown here to be rounded, affording a small space between the valve and the seat. These spaces allow good flow characteristics and

freedom from mechanical interference (e.g. preventing corners from interfering). Additionally, these spaces are available to tolerate an occasional crumb of carbon or cinder without damage.

Figures 9 A, B, C depict three embodiments of the present invention wherein the substantially cylindrical portion of the valve head having different depths or axial lengths as the substantially cylindrical mating surface formed into the port. As shown in **Figure 9A**, valve head comprises substantially cylindrical portion 902 and conical portion 904. Cylindrical portion 902 slideably couples its corresponding cylindrical mating surface 906. Conical mating surface 908 and substantially cylindrical mating surface 906 are formed in port opening so that corresponding cylindrical portion 902 and conical portion 904 fit when valve is properly seated. It should be noticed that the depth of substantially cylindrical portion 902 is much less than the depth of substantially cylindrical mating surface 906. Moreover, for purposes of the present invention, substantially cylindrical portion 902 may be only a sliver of material. It suffices for the purposes of the present invention that, as the poppet valve is moving axially down from its fully seated position, a sufficient seal is maintained between the substantially cylindrical portion 902 and its corresponding cylindrical mating surface 906. Thus, gases entering or exiting through the valve do not see an effective opening until the substantially cylindrical portion 902 passes the bottom end of the substantially cylindrical mating surface 906. By that time, the poppet valve will be traveling with sufficient velocity for the gases to experience a very sudden opening.

Figure 9B depicts an additional embodiment wherein a conical portion 904 is in between two substantially cylindrical portions 902 and 910. It should be noticed that cylindrical portion 902 has much less depth than its corresponding cylindrical mating surface. In fact, one or both of the cylindrical portions could be smaller in axial length than its corresponding cylindrical mating surface. **Figure 9C** depicts one such embodiment wherein two cylindrical portions 902 are smaller in axial length than their corresponding cylindrical mating surfaces 906.

Figure 10 depicts a comparison between a conventional poppet valve opening characteristic curve 1002 versus a poppet valve made in accordance with the

principles of the present invention (curves 1004 and 1006). Curve 1002 shows the valve lift and duration of lift for a conventional poppet valve. For curve 1002, the valve lift is 0.40 inches and the duration of time where the valve is lifted from its seat is 240 degrees of crankshaft travel.

For purposes of the present invention. "total duration" is the amount of time (measured in crankshaft degrees) that the valve is lifted off its seat. "Effective duration" is the amount of time (measured in crankshaft degrees) that the valve is lifted off its seat and gases see an effective opening at the valve. Thus, for conventional poppet valves, effective duration equals total duration. Additionally, "total lift" is the amount of lift (or axial displacement) of the valve at any given time. "Effective lift" is the amount of lift of the valve minus the amount of lift required for the valve to travel in order for gases to see an effective opening. Again, for conventional poppet valves, total lift equals effective lift.

Curve 1006 is a curve of total lift versus total duration for a poppet valve made in accordance with the present invention. As can be seen, the amount of total lift is greater for the presently claimed poppet valve system because its cam is designed to provide greater lift for advantages that will be discussed below. Additionally, total duration for the presently claimed poppet valve system is also longer, as the profile of the cam is designed to provide this duration. It is an easy matter to redesign the lobe of the cam (i.e. its lifting portion) to be higher and broader to provide this level of total lift and total duration. Additionally, since the lobe can be made broader as well as higher, the slope of its rise can be made to match that of a conventional cam acting on a conventional poppet valve. Thus, the accelerations on the valve mechanism are the same as in the conventional case -- providing an acceptably low amount of wear and stress on the valve system.

Now, curve 1004 depicts the effective lift of the presently claimed poppet valve system versus its effective duration. Because gases do not see an effective opening until after a certain amount of lift and duration, the effect is shown as a downward translation of curve 1006. Several advantages are now seen as a result. First, the presently claimed poppet valve system has a shorter open duration than the conventional poppet valve system. Second, the valve has about a nine percent

increase in breathing capacity than the conventional poppet valve -- as breathing is proportional to the area under the curve.

The total lift of the presently claimed poppet valve in this example is 0.52 inches, taking place in a total duration of 312 degrees. However, the effective lift is 0.414 inches and the effective duration is 200 degrees which is only 83 percent of the duration of the conventional poppet valve. This combination will yield a more tractable and appropriate automobile engine having the smooth slow idle (less fuel consumed in stop and go traffic); greater pullup torque (easier acceleration with less changing of transmission gear ratios); a broader power curve (less need to race the engine when passing or ascending hills); and improved fuel economy resulting in lower emissions.

The increase in total lift of the presently claimed poppet valve is offset by the increase in total duration so that acceleration loads on the mechanism are the same as in the conventional poppet valve system.

In order for the valve spring to cope with the greater foreshortening of the spring when the cam lobe forces it down, the valve spring may be designed to be longer (30 percent in the present example). In order to accommodate this change in spring length, the valve stem itself may be made longer. Thus, in our example, if the original length of the conventional valve spring was one inch when the cam lobe compressed it fully, then the new spring (in the presently claimed poppet valve system) could be made having a compressed length of 1.30 inches and thus the valve stem would be 0.3 inches longer than the conventional valve stem.

It will be appreciated that this particular example is merely one possible lift versus duration curve. Other curves may be constructed by varying the above parameters (e.g. valve length, spring length, depth of lift before effective opening). Thus, the present invention provides a method of creating a breathing performance characteristic unique to any automotive application.

It will be appreciated that other ranges and dimensions are possible for the purposes of the present invention. For example, field experience with the valves of the present invention might indicate other suitable ranges or dimensions. Additionally, different applications might themselves indicate other suitable ranges or

dimensions. Thus, the scope of the present invention should not be limited by the recitation of ranges or dimensions and these are included to provide a possible set of working values. The scope of the present invention contemplates and encompasses all obvious extensions of the basic principle as herein set forth.

It will now be described how an efficient Miller cycle operation may be effected by use of the fast opening poppet inlet valves. as described above.

Figure 11A-D depict the inlet valve timing for Miller cycle operation. **Figure 11A** shows conventional operation of an internal combustion engine using conventional poppet valves. In this example. inlet opening ("IO") occurs before Top Dead Center (bTDC) until inlet closing ("IC") after Bottom Dead Center (aBDC). It should be appreciated that, with conventional poppet valves. IO occurs as the cam is forcing the conventional poppet valve off of its seat. because there is no effective sealing of the conventional poppet valve during any portion of its travel. This Figure depicts that amount of crankshaft angle of effective opening, in this case, is 240 degrees. with the dotted line indicating the midpoint of opening travel.

As IO occurs bTDC while IC occurs aBDC. this relatively long open duration is necessary with conventional poppet valves because their initial opening and final closing takes place gradually. Thus. in order to obtain good cylinder charging (i.e. high volumetric efficiency), the total open duration must be extended out to the point where it begins before and ends after the entire inlet stroke. In the Miller cycle engine. the inlet stroke is curtailed either by early inlet closing or by extra late inlet closing in order to limit the volume of inlet air or fuel vapor. It could be said that the volumetric efficiency is therefore low in the Miller cycle.

However. as will be shown below in discussion of the Miller cycle operation using fast opening poppet inlet valves of the present invention. a less-than-complete inlet stroke is followed by a compression stroke that forces the smaller-than-normal fuel-air charge into a smaller-than-normal combustion chamber. This has two principal benefits – first: it increases T1. the high temperature at the beginning of the expansion process; and second. it increases the expansion ratio so that T2 (the temperature of the gases at the end of the expansion when the exhaust valves opens)

is below normal. Thus, in keeping with Carnot's equation, (T1-T2)/T1, the amount of work done during expansion process is greater relative to the total heat available.

By "combustion chamber", it is meant that volume in the cylinder head when the piston is at TDC where combustion is initiated. It is often considered to be the "minimum cylinder volume" whereby the "maximum cylinder volume" is the minimum cylinder volume plus the volume in the cylinder with the piston at BDC. The compression ratio is the maximum volume divided by the minimum volume. Thus, if the combustion chamber volume is small relative to the cylinder volume, the compression ration will be high. The expansion ratio is the volume at the beginning of the expansion process divided into the volume at the end of said process when the exhaust valve opens and the gases are released. The expansion ratio is usually slightly less than the compression ration, since it begins with the minimum cylinder volume but ends slightly before the piston is all the way down when the exhaust valve opens. The two ratios are thus closely related and are similar but not exactly the same in numeric value.

Figures 11B to 11D shows one embodiment of valve timing diagrams of the fast opening poppet inlet valves of the present invention effected by the mechanism of **Figure 13** when read in conjunction with **Figures 14 A, B and C**, respectively. **Figure 11B** depicts a valve timing diagram of the fast opening poppet inlet valve during idle or light load engine operation. As noted, there is a time indicated when the fast opening poppet inlet valve "leaves its seat" ("LS") and at a later time when there is an effective opening of the port ("IO"). IO occurs, in this example, sometime aTDC. The port opening continues for some specified crankshaft angle (in this figure, 70 degrees) until the inlet port closes ("IC"), then, later, the fast opening poppet "returns to its seat" ("RS").

In **Figure 12**, the opening curve that corresponds to this timing schedule is the curve of least amplitude 1202. As shown there, there is a horizontal line 1208 demarking where the port "sees" an effective opening and closing, depending on whether the curve has positive or negative slope, respectively. Thus, for curve 1202, the inlet sees an effective opening at about 10 degrees aTDC and an effective closing at about 90 bBDC. The lift of the valve on curve 1202 is about .12 inches. Since this

valve opens and closes abruptly as is characteristic of fast opening poppet valves – rather than gradually, as with conventional poppet valves, throttle losses through the valve itself are lower than for conventional poppets.

As will be discussed in connection with the linkage mechanism of **Figure 13, Figure 14A** depicts the setting of the adjustable linkage mechanism that corresponds to idle or low power engine operation. As noted, the mechanism is set so that the point of valve travel is at its relative minimum setting – to produce the lift as just mentioned.

Figure 11C depicts an exemplary timing schedule of the fast open poppet inlet valve during high power engine operation at sea level. As can be noted, the amount of crankshaft angle for which the inlet port sees an effective opening is greater than for idle or low power engine operation. Curve 1204 in **Figure 12** corresponds with this operation; and, as is seen, the lift is also higher than for idle and low power engine operation. **Figure 14B** corresponds to the settings of the novel linkage effecting this high power engine operation at sea level.

Figure 11D corresponds to engine operation at high altitude where full aspiration of the inlet charge is more challenging. As can be seen, the valve timing is such that the effective opening is near a maximum, while still providing Miller cycle operation. Curve 1206 of **Figure 12** corresponds to this high altitude operation. As noted, the amount of lift is at a near maximum for the particular linkage as shown in **Figure 14C**. The amount of aspiration during an inlet stroke is roughly proportional to the area under the curve in **Figure 12**. Thus, at high altitude, the amount of aspiration is greatest to compensate for the lack of atmospheric pressure at altitude.

It will be appreciated that although **Figures 11B-D** depict a Miller cycle operation with inlet closing before bottom dead center, the present invention could be made to function with a Miller cycle operation that closes the inlet after bottom dead center. The scope of the present invention contemplates both types of Miller cycle timing.

Figure 13 shows the novel linkage mechanism of the present invention that is one embodiment of effecting the Miller cycle engine operation with fast opening poppet inlet valves. Cam 1308 pushes on roller follower 1310 and transmits its

translation motion to roller 1312 via link 1311. Roller 1312 in turn pushes upon the arcuate surface of rocker 1314. Rocker 1314 is rotatably coupled to the head at fulcrum 1316. Depending upon where roller 1312 is contacting with rocker 1314, the amount and duration of lift dynamically varies. Another advantageous feature of the mechanism as shown is that as control eccentric 1320 is moved clockwise, it draws cam follower 1310 and roller 1312 to the left, causing valve 1318 to open farther as in **Figure 14B**. Roller follower 1310 moves clockwise about center of cam 1308, causing average open time (represented by dotted lines in **Figure 11** to be later relative to crankshaft position). This effect may be noted in **Figure 11B, Figure 11C,** and **Figure 11D**. The effect is also responsible for the progressive shift to the right in **Figure 12** of curves 1202, 1204, and 1206.

Link 1311 is connected to link 1315 via tie links 1313. Link 1315 is connected to shifting eccentric 1320 which is rotatably coupled to the head at fixed center 1322. The speed and power control of the engine is effected by shifting eccentric 1320. Shifting this eccentric corresponds to moving a throttle valve of a conventional engine: but without an actual throttle and its attendant losses.

It will be appreciated that the linkage mechanism of **Figure 13** could also be applied to conventional poppet valve engines for augmenting performance by providing a more continuous range of valve timings -- not limited to two or more timings; but allowing any timing to optimize performance under the prevailing driving conditions.

An alternative embodiment is depicted in **Figure 15**. The control of the poppet valve may be effected by actuator 1502.. It is well known in the art to make a suitable actuator to control both the duration of opening and amount of lift by electrical, electromagnetic, solenoid, pneumatic, or hydraulic actuators, or any combination thereof. Such actuator 1502 can additionally be controlled under microcontroller 1504 that senses both the power demand of the driver and the ambient atmospheric pressure environment to determine the optimum operation of the inlet poppet valve.

Figure 16 depicts one embodiment of a control system for the fast opening poppet inlet valve system to effect an efficient Miller cycle operation. Accelerator

pedal 1602 is linked to pedal position sensor 1604 which gives data to electronic control unit 1622. ECU 1622 may be implemented by a microcontroller under program control, as is well known in the art. Also, giving data to ECU 1622 are barometric pressure sensor 1606 (to measure ambient atmospheric pressure as a driving condition parameter), knock sensor 1608, ignition timing input 1610, fuel injection input 1614, engine temperature input 1614, engine speed input 1616, and ignition and fuel injection delay circuit 1628.

These sensor-control systems are advantageous to the operation of the engine because the engine, in order to optimize the Miller cycle operation, has a relatively high compression ratio. The method by which detonation (i.e. knocking) is avoided in such an engine is to limit the quantity of inlet air per inlet stroke.

As the inlet valve opening may be kept small and of short open duration at sea level barometric pressure, there is an option open to maintain sea level power output at very high altitudes by way of a more normal inlet valve opening and duration under those conditions.

However, at more moderate altitudes of higher barometric pressure, the speed control must never be opened up to such an extent, lest violent detonation occur. Thus, if the driver wanting maximum power, presses the accelerator to the floor, the barometric pressure sensor 1606 must intercede, and along with air temperature sensing (warm air aggravates detonation) must limit the amount that power control servo 1618 "opens up". If ever detonation does begin, even in small degree, detonation sensor 1608 (sometimes called a "knock" sensor) will command the power servo 1618 to back off a little. This feature also comes into play in the event that low grade fuel of poor anti-knock qualities is used.

When a four-stroke internal combustion engine is stopped, it will come to rest with one or more inlet valves open. The cylinders where these valves are open will fill with air at atmospheric pressure. If this engine is started with fuel injection and ignition functioning, then a cylinder filled completely with air (as opposed to the normal inlet restriction by valve control) may experience a damaging detonation the first time through.

The engine ignition and fuel injection delay sensor 1628 prevents this by responding to input from the engine speed sensor 1616. Sensor 1628 simply prevents ignition or fuel injection until the engine has reached a moderate cranking speed, perhaps around 150 RPM(although other speeds may work as well). This assures that all cylinders are cleared of atmospheric air and are now under the influence of inlet air restriction via the Miller cycle scheme.

Ignition timing 1610 is sensitive to engine speed 1616, advancing the spark at higher speeds and at light load, retarding it for heavy load or whenever knock sensor 1608 senses detonation, and also when idling or starting.

Fuel injection sensor 1612 moderates the duration of each injection shot of fuel into the inlet airstream. It is sensitive to oxygen sensor 1630 and limits the fuel injection to the point where a specified oxygen concentration appears in the exhaust gasses thus maintaining the fuel-air ratio just on the lean side of stoichiometric. This minimizes unburned hydrocarbons and improves fuel economy.

Engine temperature sensor 1614 monitors the temperature of the engine coolant. When the temperature of the engine coolant is cold, as when first starting, it lengthens the fuel injection duration, thus enriching the fuel-air ratio. In some systems, this monitor will open a priming fuel injector while the starter is engaged and the engine cold. When the engine warms up, it can moderate exhaust gas recirculation in those engines so equipped. It can start and stop fans for the radiator, thus conserving fan power when not needed.

Sensor 1624 controls an exhaust gas recirculation valve. It senses engine load as a function of engine speed 1616 and position of servo 1618. It causes some amount of the exhaust gasses to be fed back into the air inlet. Its principle function is to limit flame temperature by displacing some of the oxygen with carbon dioxide, thus reducing the tendency to detonate and reducing emissions of oxides of nitrogen. It also reduces unburned hydrocarbons by running them through a second time.

Sensor 1626 measures inlet air flow rate and makes the primary modulating control to the fuel injection shot duration so as to rough out the quantity of fuel entering the inlet air to be further fine tuned by the other sensor parameters.

All the input information is processed in ECU 1622 which selects the most appropriate power setting in the valve control mechanism 1620 and actuates the mechanism 1620 via a power control servo 1618. It will be appreciated that, although **Figure 16** shows only the mechanical actuation of the fast opening poppet inlet valve, it would be obvious to one skilled in the art to use an alternative electromagnetic, solenoid, pneumatic, hydraulic, or some other suitable actuation system in lieu of the mechanical actuator as shown here. The scope of the present invention encompasses all such alternative embodiments.

One advantageous feature of this control system is altitude compensation. The barometric pressure sensor can cause the servo 1618 under ECU control to allow the inlet valves to let in nearly twice the air at high altitudes as they do at sea level. This may also be accomplished without need for supercharging in the inlet air/fuel mixture. This results in a uniform availability of maximum power regardless of altitude. It is also possible to measure the ambient air temperature to fine tune the amount of lift the inlet valves experience.

It will be appreciated that any subset of sensor and control parameters may be implemented on any particular engine design, with varying degrees of performance results. The present invention contemplates the use of none, some or all of the aforementioned sensors in conjunction with a fast opening poppet inlet valve. The present invention additionally encompasses all obvious variations of the foregoing.

Claims

1. In a valve system comprising a valve that fits an aperture in a housing, said aperture comprising a substantially cylindrical mating surface, said valve system comprising:

a valve stem;

a valve head coupled to said valve stem, wherein said valve head further comprises a substantially cylindrical portion; and

wherein said valve head moves within said housing such that said substantially cylindrical portion provides an effective seal with said substantially cylindrical mating surface of said aperture.

2. The valve system as recited in Claim 1 whereby said aperture further comprises a conical mating surface and a substantially cylindrical mating surface;

said valve head further comprises a conical portion and a substantially cylindrical portion;

wherein said valve head slideably couples with said housing such that said conical portion couples with said conical mating surface and said substantially cylindrical portion couples with said substantially cylindrical mating surface.

3. The valve system as recited in Claim 2 whereby said conical portion couples with said valve stem and said substantially cylindrical portion couples with said conical portion.

4. The valve system as recited in Claim 2 whereby said substantially cylindrical portion couples with said valve stem and said conical portion couples with said substantially cylindrical portion.

5. The valve system as recited in Claim 2 whereby said valve head further comprises a conical portion, a first substantially cylindrical portion and a second substantially cylindrical portion;

whereby said valve stem couples with said first substantially cylindrical portion, said first substantially cylindrical portion couples with said conical portion, and said conical portion couples with said second substantially cylindrical portion.

6. The valve system as recited in Claim 1 wherein said substantially cylindrical portion has a depth less than its corresponding substantially cylindrical mating surface.

7. The valve system as recited in Claim 1 wherein said valve system regulates the flow of gases through said housing; and

wherein an effective opening for the flow of said gases occurs when said substantially cylindrical portion moves past the end of said substantially cylindrical mating surface.

8. The valve system as recited in Claim 1 wherein the depth of said substantially cylindrical mating surface is at least 4 percent of the valve lift.

9. The valve system as recited in Claim 1 wherein the depth of said substantially cylindrical mating surface is at least 10 percent of the valve lift.

10. The valve system as recited in Claim 2 wherein said valve head further comprises a relief groove.

11. The valve system as recited in Claim 2 wherein said valve head further comprises a gas trapping groove.

12. The valve system as recited in Claim 2 wherein the length of said conical portion is at least 4 percent of valve head diameter.

13. The valve system as recited in Claim 2 wherein the length of said conical portion is at least 6 percent of valve head diameter.

14. The valve system as recited in Claim 2 wherein said conical portion further comprises rounded corners.

15. In an internal combustion engine comprising at least one cylinder comprising an inlet and exhaust port. wherein at least one of said inlet and exhaust ports is activated by a poppet valve comprising a valve stem and a valve head coupled to said valve stem;

an improved poppet valve system wherein said improvement comprises:

a valve stem;

a valve head coupled to said valve stem wherein said valve head further comprises a substantially cylindrical portion;

wherein said at least one of said inlet and exhaust ports comprises a substantially cylindrical mating surface; and

further wherein said substantially cylindrical portion of said valve head moves along said substantially cylindrical mating surface of said at least one of said inlet and exhaust ports.

16. The improved poppet valve as recited in Claim 15 whereby said valve head further comprises a conical portion and a substantially cylindrical portion.

17. The improved poppet valve as recited in Claim 16 whereby said substantially cylindrical portion and said conical portion of said valve

head slideably couples to at least one of said inlet and exhaust ports comprising a substantially cylindrical mating surface and conical mating surface.

18. The improved poppet valve system as recited in Claim 15 wherein said substantially cylindrical portion has a depth less than its corresponding substantially cylindrical mating surface.

19. The improved poppet valve system as recited in Claim 15 wherein gases flows through said improved poppet valve system only when said substantially cylindrical portion moves past the end of the said substantially cylindrical mating surface.

20. An poppet valve system, said system comprising:

a poppet valve, said poppet valve comprising an valve stem, a valve head coupled valve stem; said valve head further comprising a substantially cylindrical portion;

a corresponding port such that said poppet valve axially displaces within said port, said port defining a substantially cylindrical surface, such that said substantially cylindrical portion axially moves within said substantially cylindrical surface;

a cam actuated system comprising a cam, a cam follower, a valve spring such that said cam actuated system forces said poppet valve to axially displace within said port; and

wherein gases regulated by said poppet valve system do not see an effective opening until said substantially cylindrical portion has axially displaced beyond the end of said substantially cylindrical surface.

21. The poppet valve system as recited in Claim 20 wherein said valve system has an effective duration less than its total duration.

22. The poppet valve system as recited in Claim 21 wherein said valve system has an effective lift less than its total lift.

23. In an internal combustion engine, a system for effecting a Miller cycle operation, said system comprising:

a head having a inlet aperture, said aperture comprising a substantially cylindrical mating surface;

an inlet poppet valve, said poppet valve comprising a valve head further comprising a substantially cylindrical portion, said substantially cylindrical portion providing an effective seal with said substantially cylindrical mating surface of said aperture;

an actuator, said actuator coupled to said inlet poppet valve for actuating the opening and closing of said inlet poppet valve such that a Miller cycle operation is effected by said actuator.

24. The system as recited in Claim 23 wherein said actuator is an electromechanical actuator.

25. The system as recited in Claim 23 wherein said actuator is an pneumatic actuator.

26. The system as recited in Claim 23 wherein said actuator is an hydraulic actuator.

27. The system as recited in Claim 23 wherein said actuator is a solenoid actuator.

28. The system as recited in Claim 23 wherein said actuator causes the said inlet poppet valve to close before bottom dead center.

29. The system as recited in Claim 23 wherein said actuator causes the said inlet poppet valve to close after bottom dead center.

30. The system as recited in Claim 23 wherein said system further comprises an electronic control unit coupled to and controlling said actuator;

wherein said electronic control unit causes said actuator to vary the opening and closing characteristics of said inlet poppet valve.

31. The system as recited in Claim 30 wherein said system further comprises a plurality of sensors coupled to said electronic control unit;

wherein said plurality of sensors detect predetermined driving conditions and send signals corresponding to said driving condition to said electronic control unit.

32. The system as recited in Claim 31 wherein one of said plurality of sensors is a barometric pressure sensor;

wherein said electronic control unit causes said actuator to vary the length of said poppet valve lift in accordance with signals received from said barometric pressure sensor.

33. The system as recited in Claim 31 wherein one of said plurality of sensors is an accelerator pedal position sensor;

wherein said electronic control unit causes said actuator to vary the length of said poppet valve lift in accordance with signals received from said accelerator pedal position sensor.

34. The system as recited in Claim 31 wherein two of said plurality of sensors is a barometric pressure sensor and an accelerator pedal position sensor;

wherein said electronic control unit causes said actuator to vary the length of said poppet valve lift in accordance with signals received from said barometric pressure sensor and said accelerator pedal position sensor;

such that when the barometric pressure sensor sends a signal of sea level pressure, said electronic control unit sets a first maximum lift position for the lift of said poppet valve according to maximum accelerator pedal position for power and;

such that when the barometric pressure sensor sends a signal of a pressure less than sea level, said electronic control unit sets a second maximum lift position for the lift of said poppet valve according to maximum accelerator pedal position for power;

wherein said second maximum lift position is greater than said first maximum lift position.

35. The system as recited in Claim 23 wherein said actuator further comprises:

a cam rotatably coupled to said head;

a roller mechanically coupled to said cam such that said roller is displaced by motion of said cam;

a rocker rotatably coupled to said head and comprising an arcuate surface; said roller coupled to said arcuate surface such that the displacement of said rocker varies according to the position at which said roller is coupled to said arcuate surface; and

a control eccentric rotatably coupled to said head and mechanically coupled to said roller such that the position at which said roller is coupled to said arcuate surface is controlled by said control eccentric.

36. The system as recited in Claim 35 wherein said rocker is coupled to said inlet poppet valve and the amount of lift of said inlet poppet valve is controlled by the position of said control eccentric.

37. The system as recited in Claim 35 wherein said rocker is coupled to said inlet poppet valve and the amount of open duration of said inlet poppet valve is controlled by the position of said control eccentric.

38. The system as recited in Claim 35 wherein said rocker is coupled to said inlet poppet valve and the midpoint of the open duration of said poppet valve varies according to the position of said control eccentric.

fig. 1
PRIOR ART

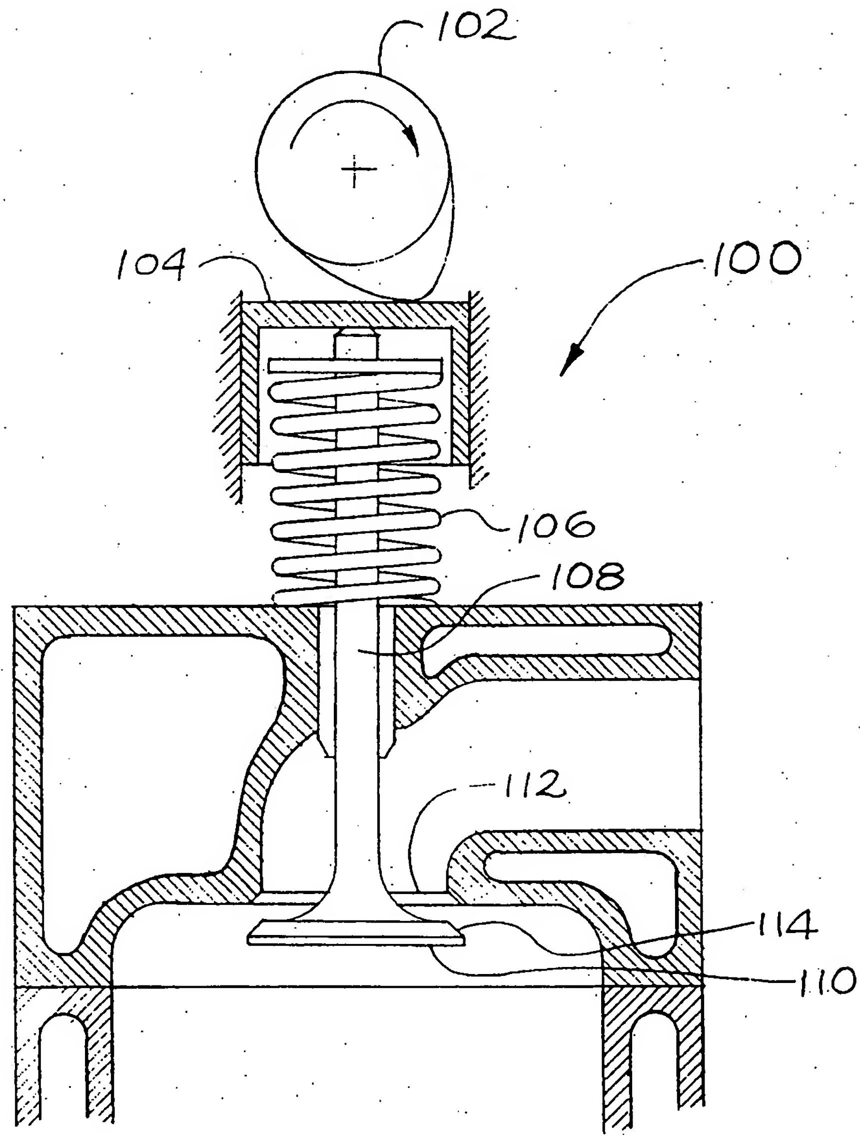


fig. 2

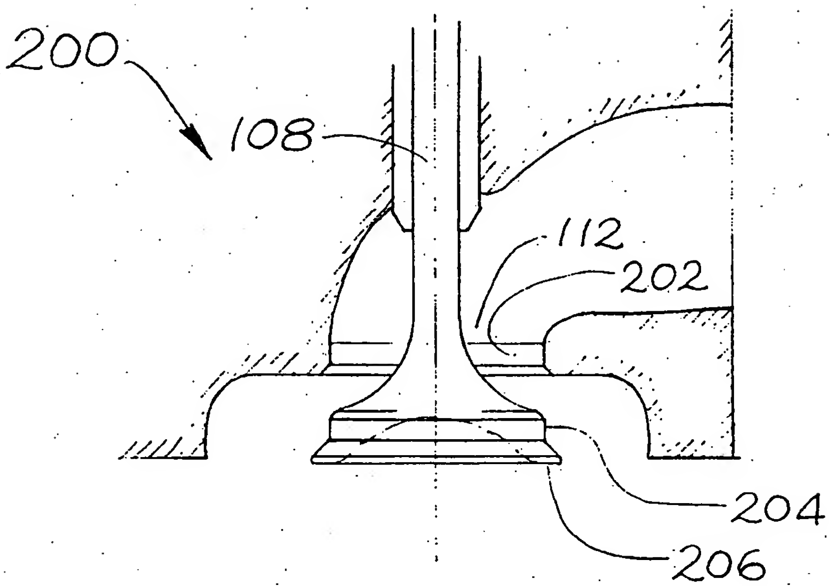


fig.3A

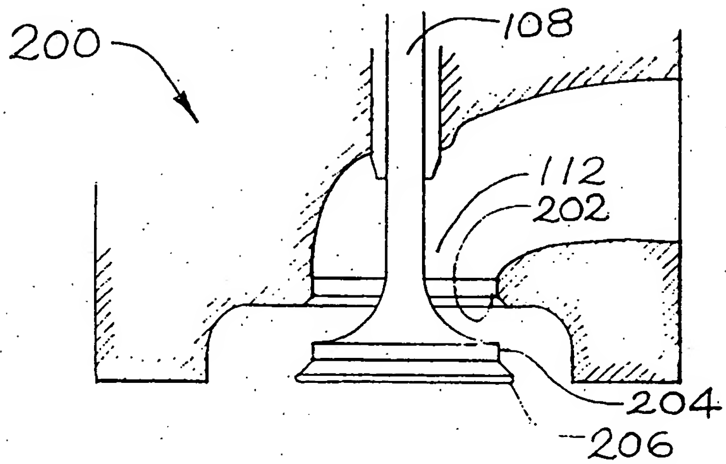


fig.3B

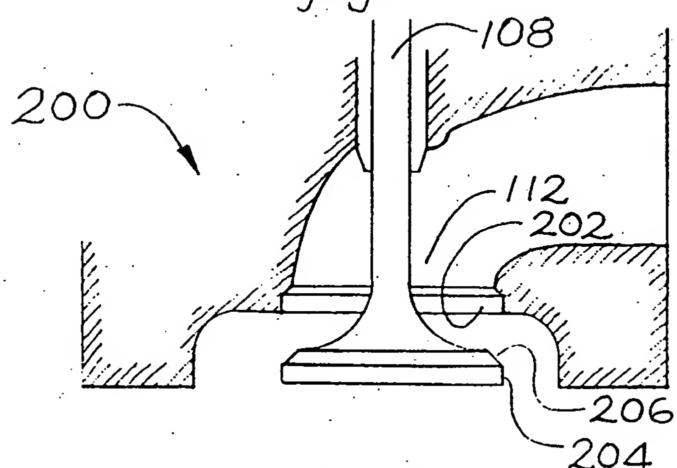


fig. 3C

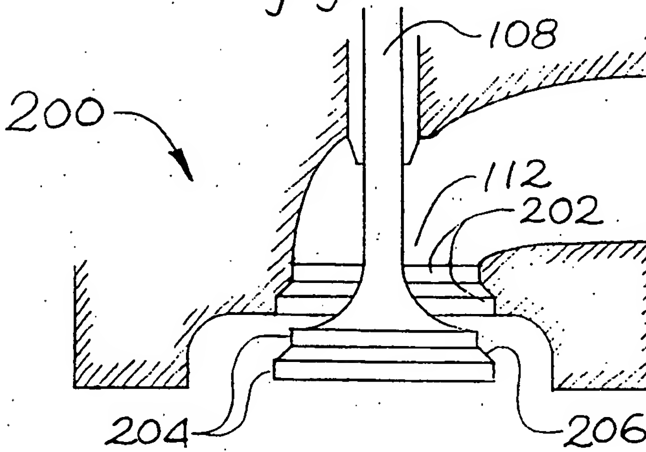


fig 4

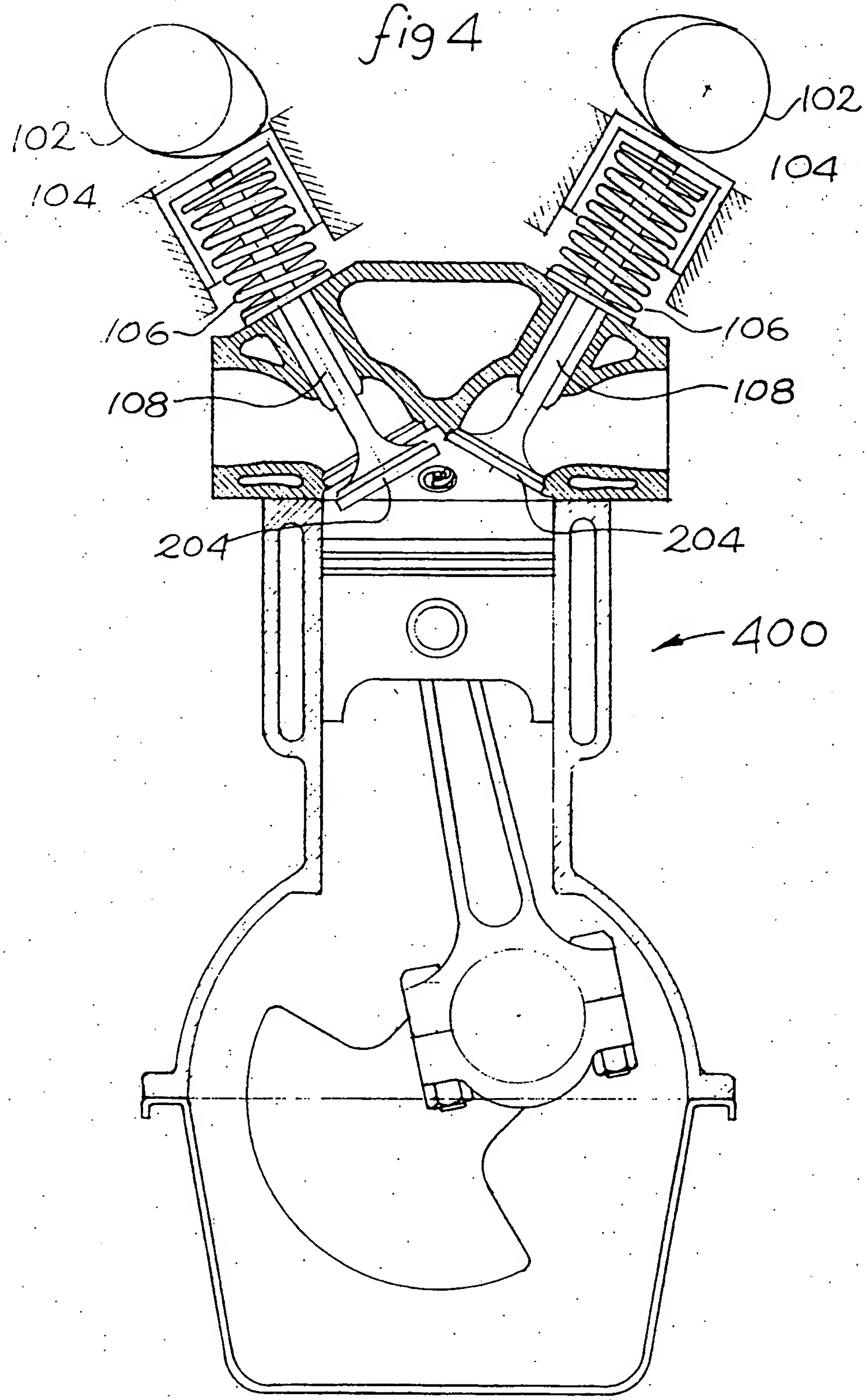
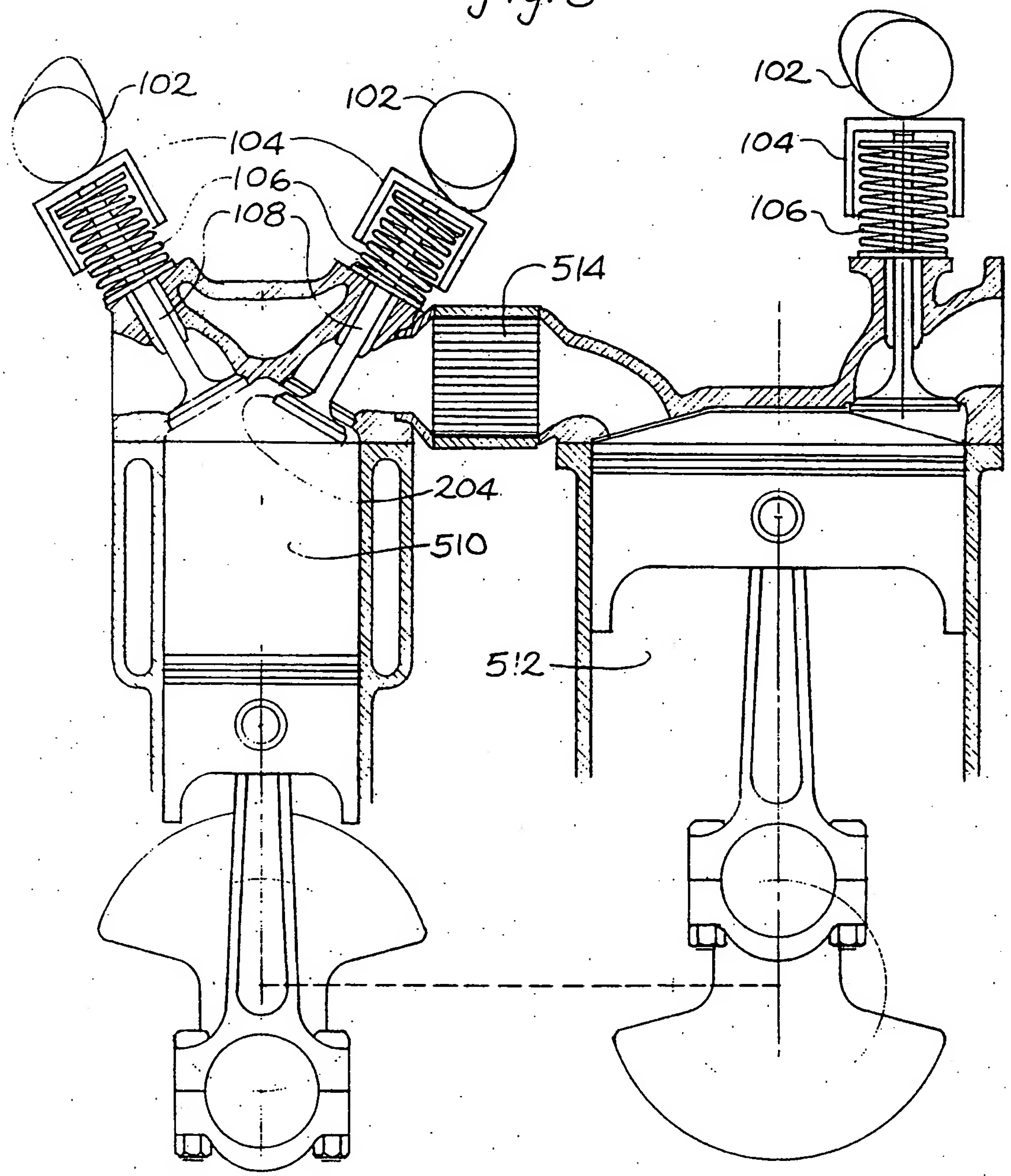


fig. 5



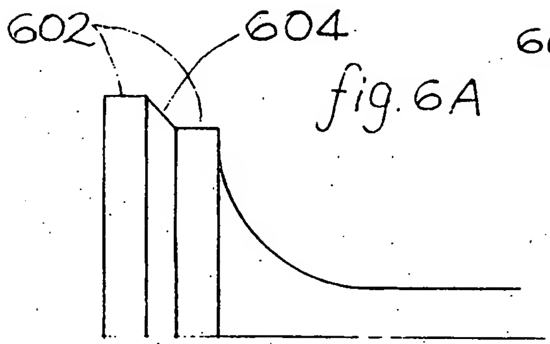


fig.6A

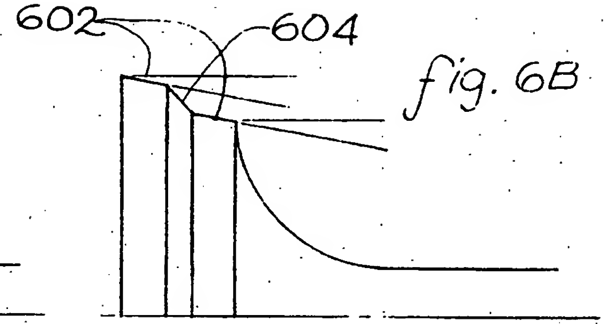


fig.6B

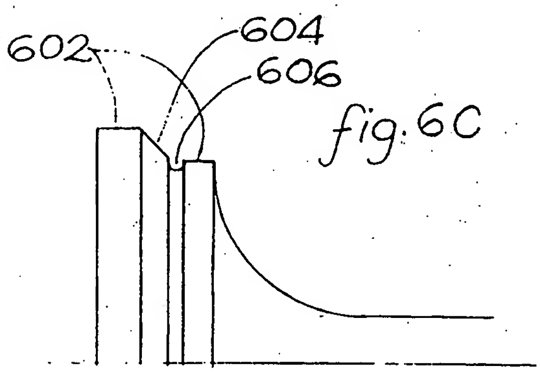


fig.6C

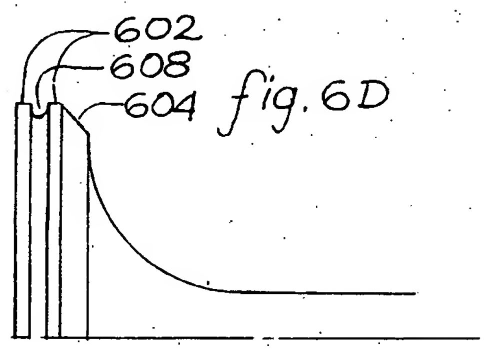


fig.6D

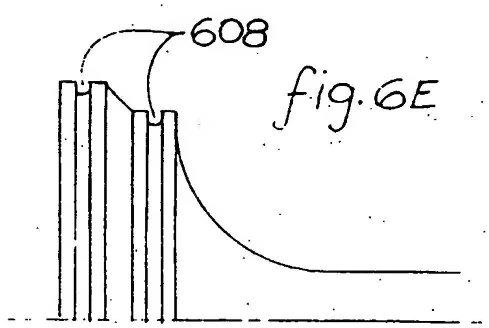


fig.6E

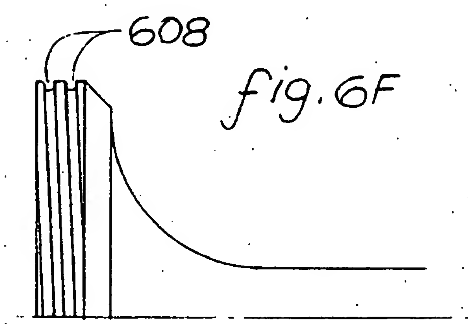


fig.6F

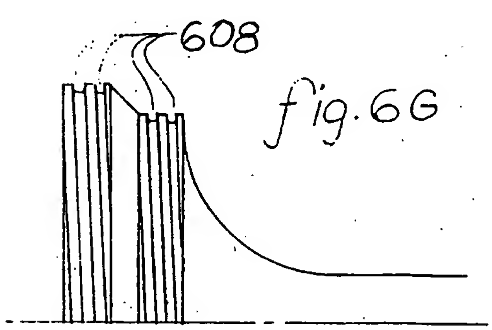


fig.6G

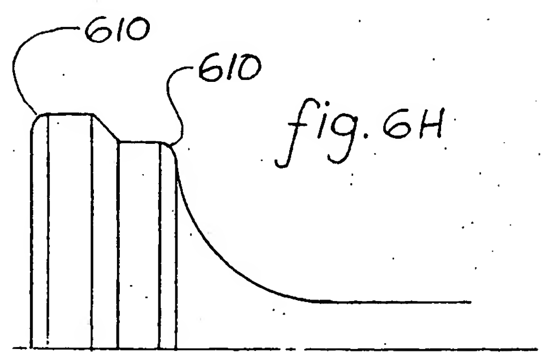


fig.6H

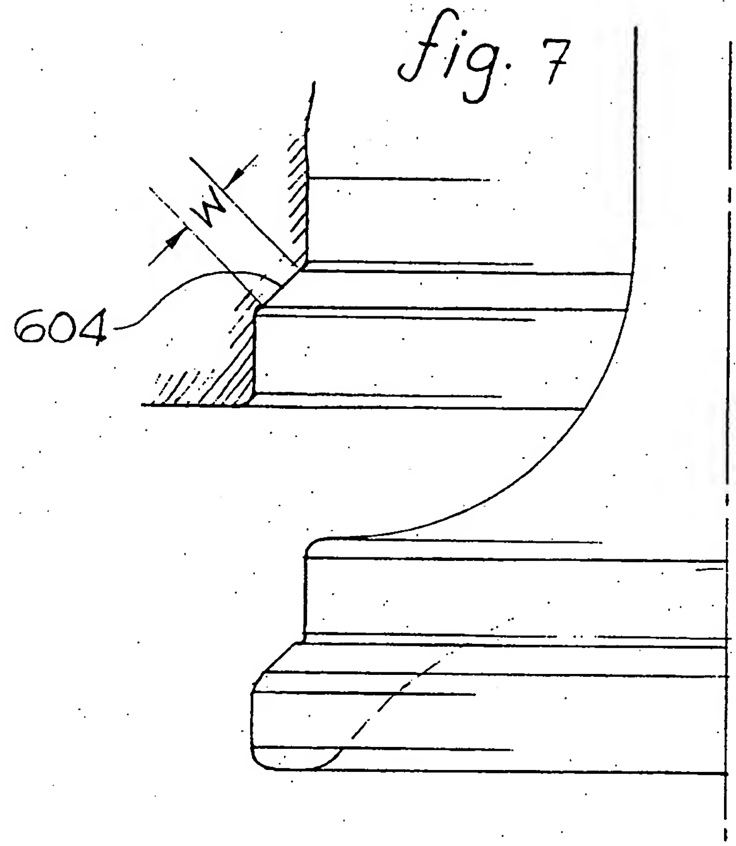


fig. 7
W
604

fig 8

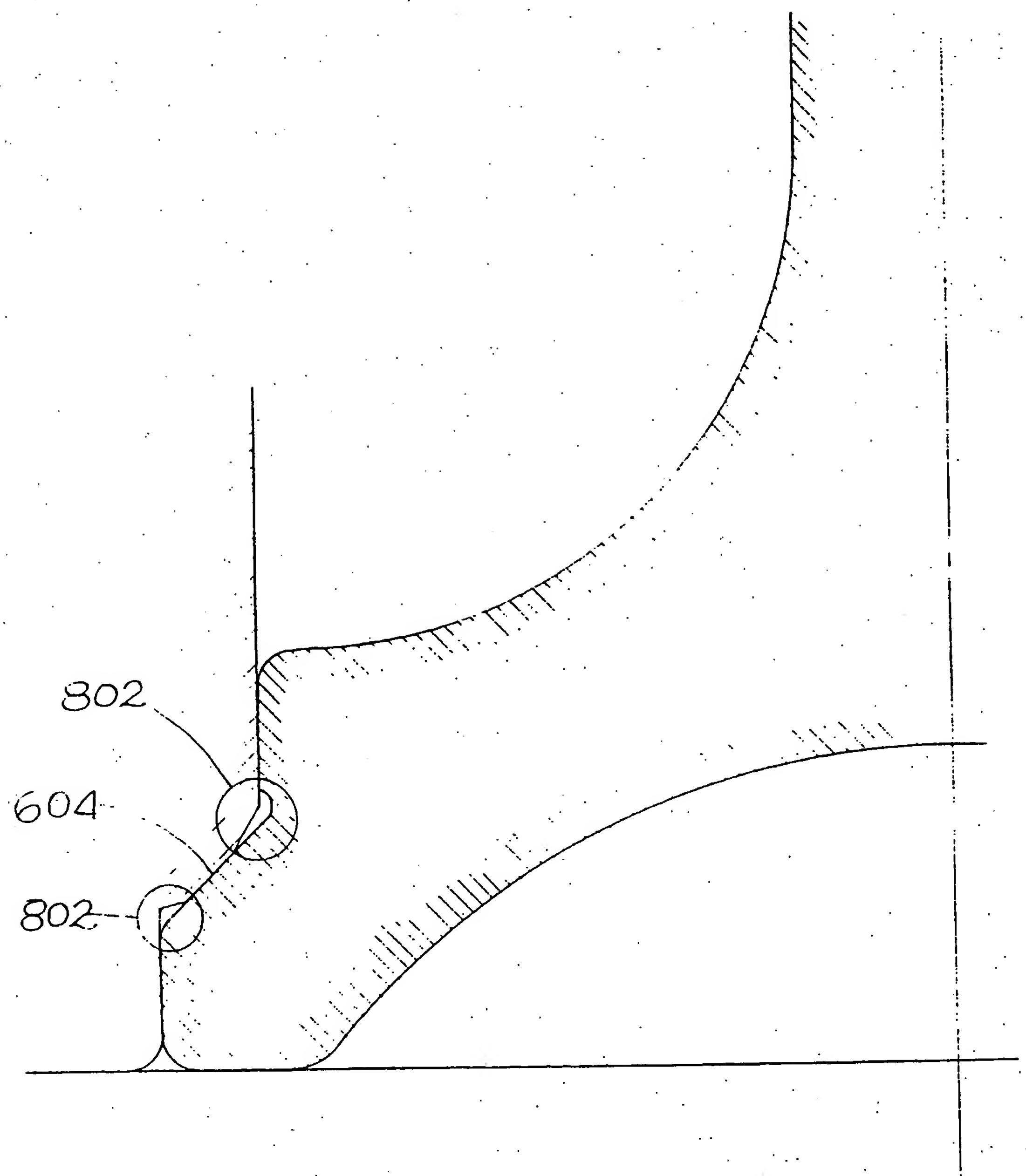


fig 9A

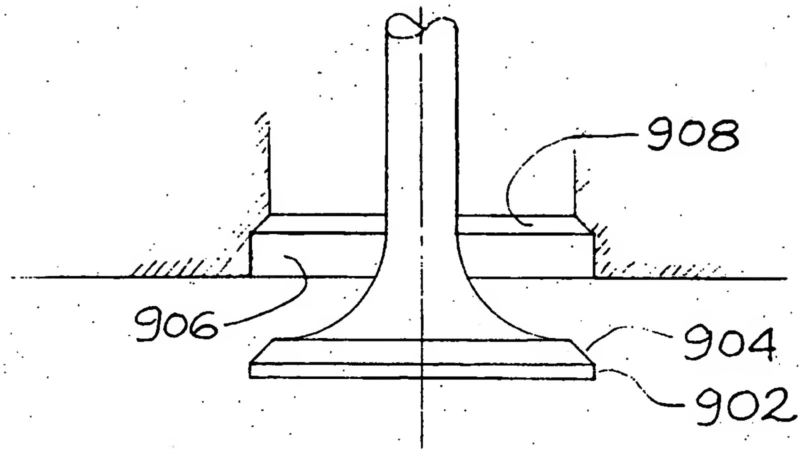


fig. 9B

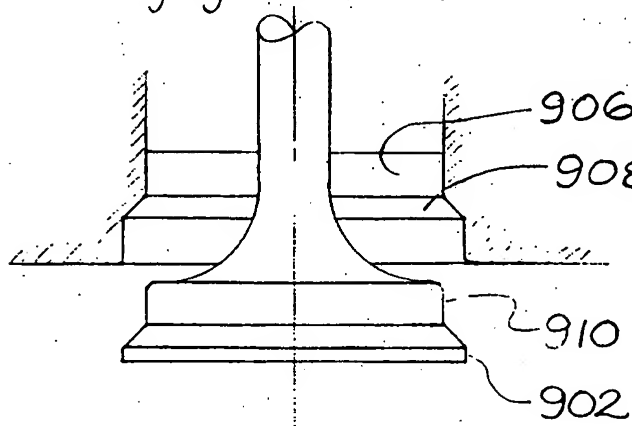


fig. 9C

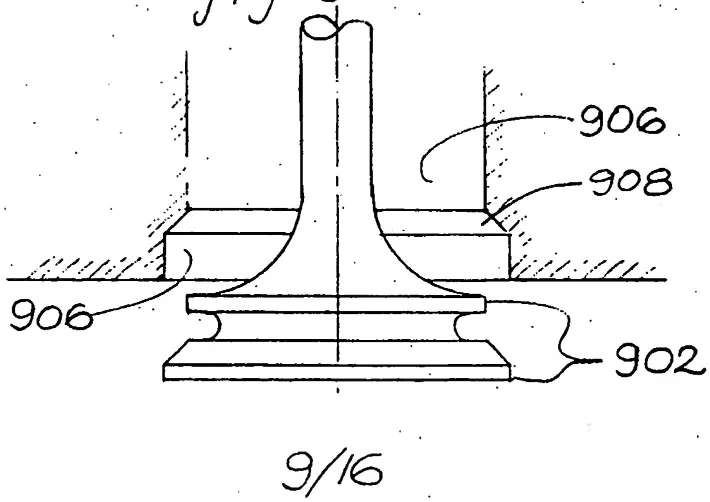
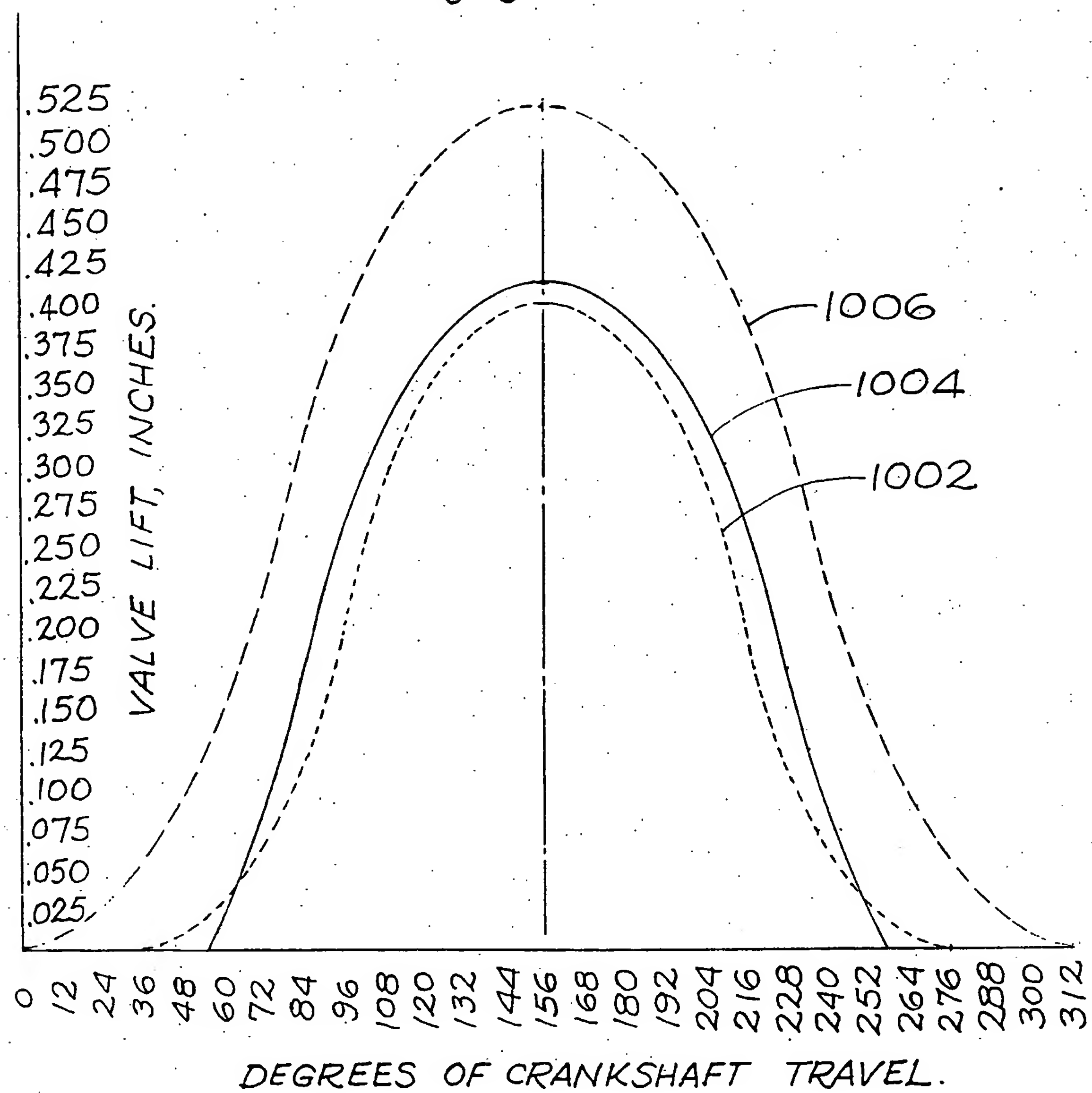


fig. 10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.100
.075
.050
.025
VALVE LIFT, INCHES.
1006
1004
1002
0
12
24
36
48
60
72
84
96
108
120
132
144
156
168
180
192
204
216
228
240
252
264
276
288
300
312
DEGREES OF CRANKSHAFT TRAVEL.

fig. 11

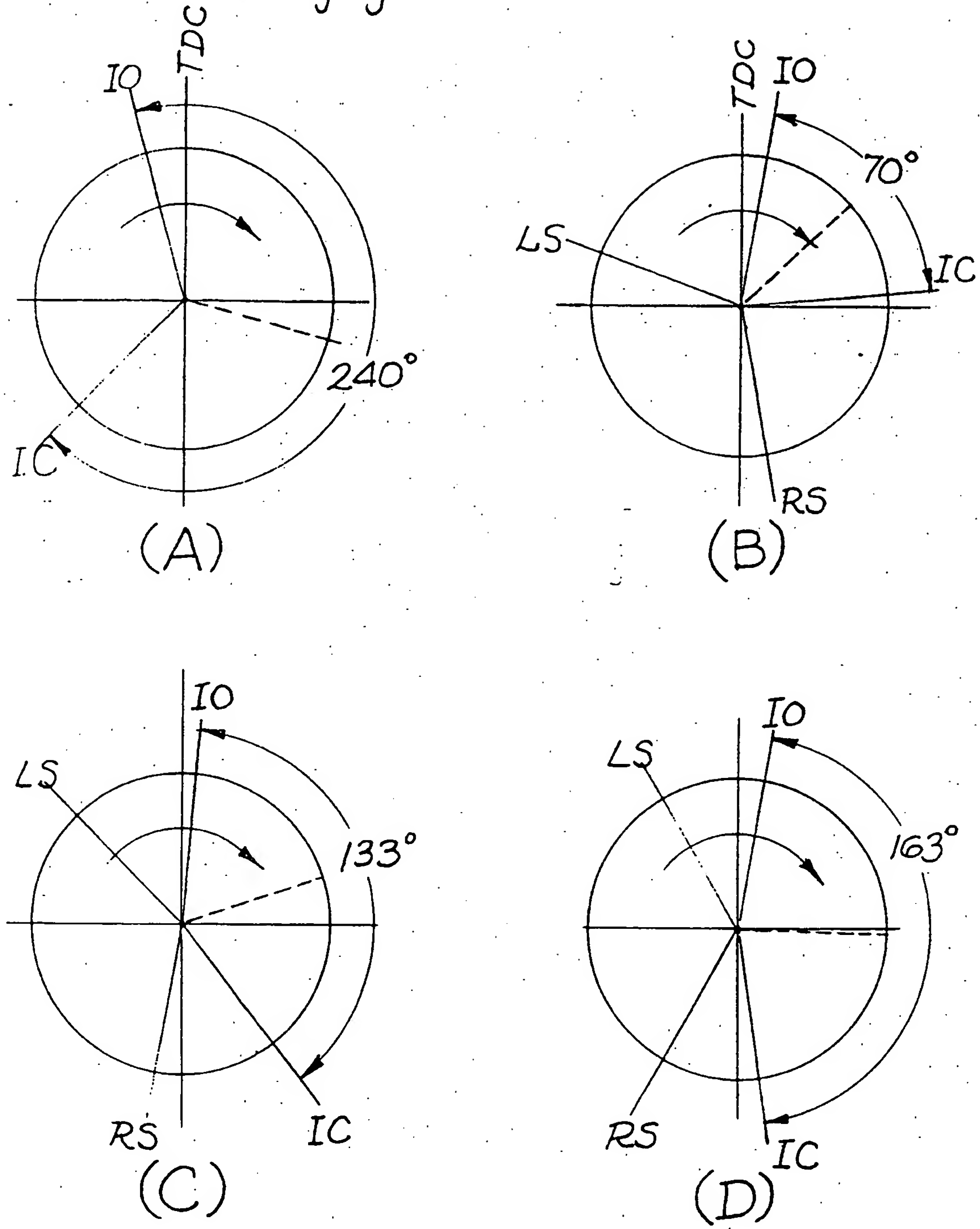
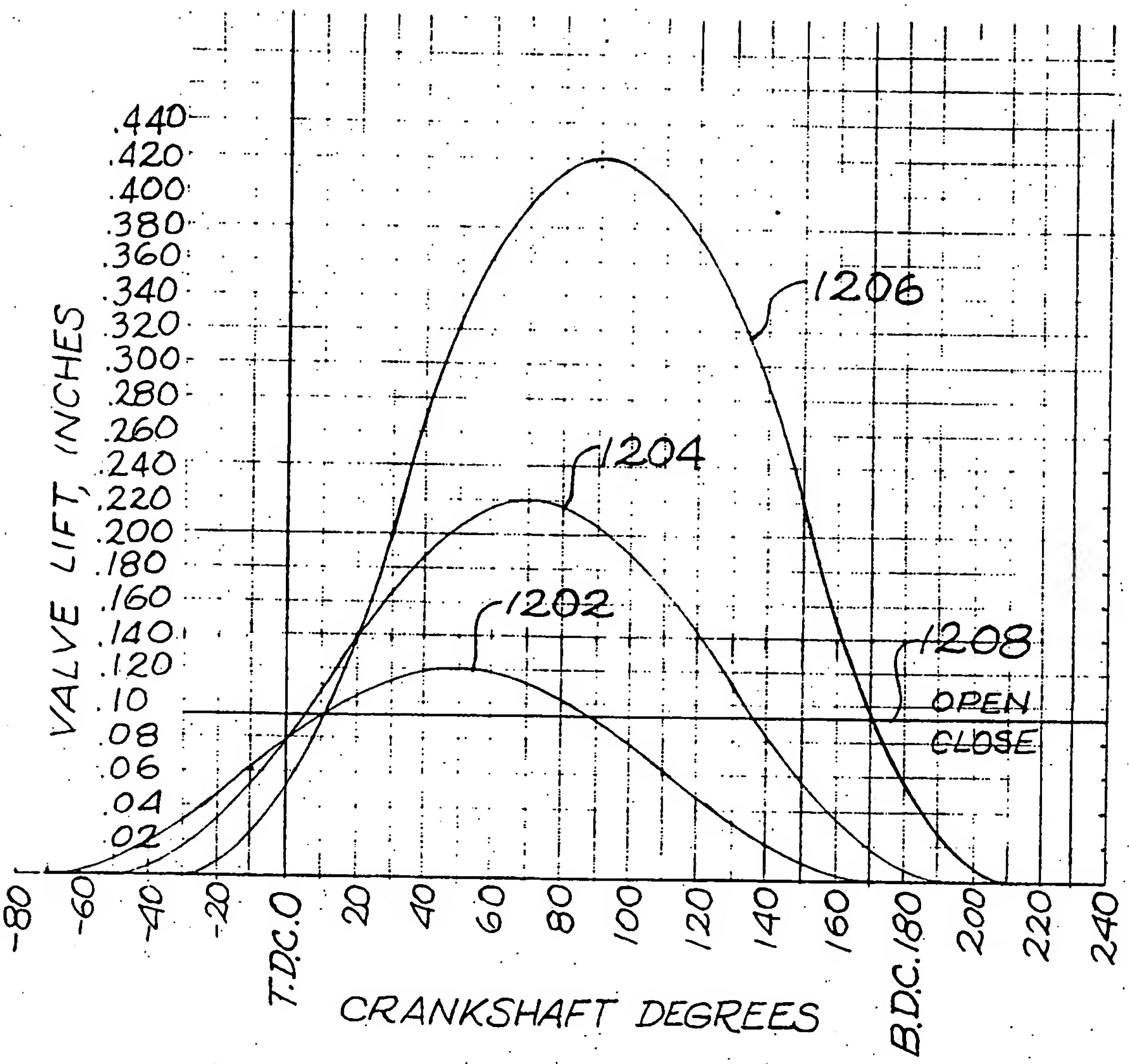


fig 12


VALVE LIFT, INCHES
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.400
.380
.360
.340
.320
.300
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.260
.240
.220
.200
.180
.160
.140
.120
.10
.08
.06
.04
.02
-80
-60
-40
-20
T.D.C.0
20
40
60
80
100
120
140
160
B.D.C.180
200
220
240
CRANKSHAFT DEGREES
1206
1204
1202
1208
OPEN
CLOSE

fig. 13

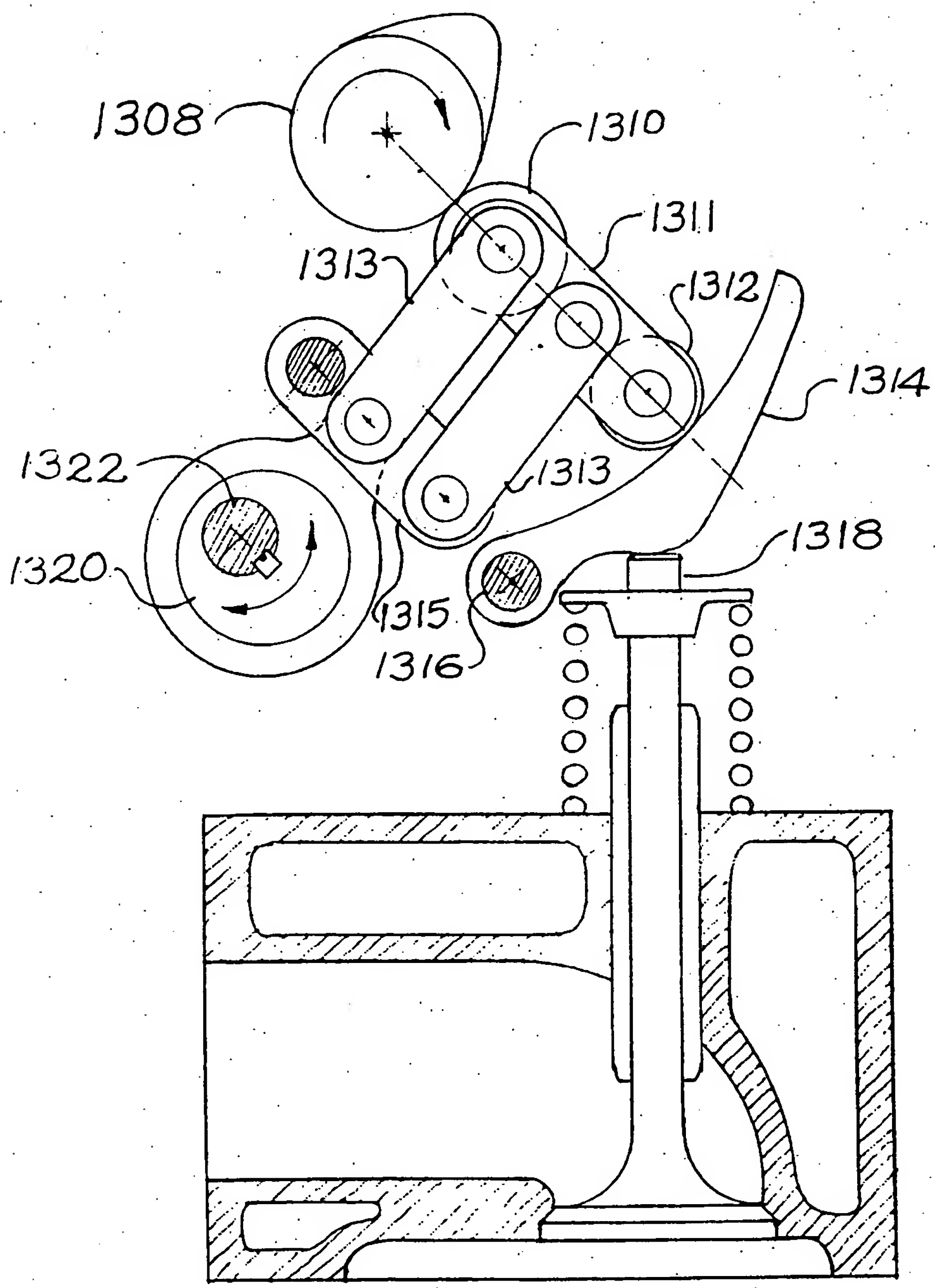


fig.14

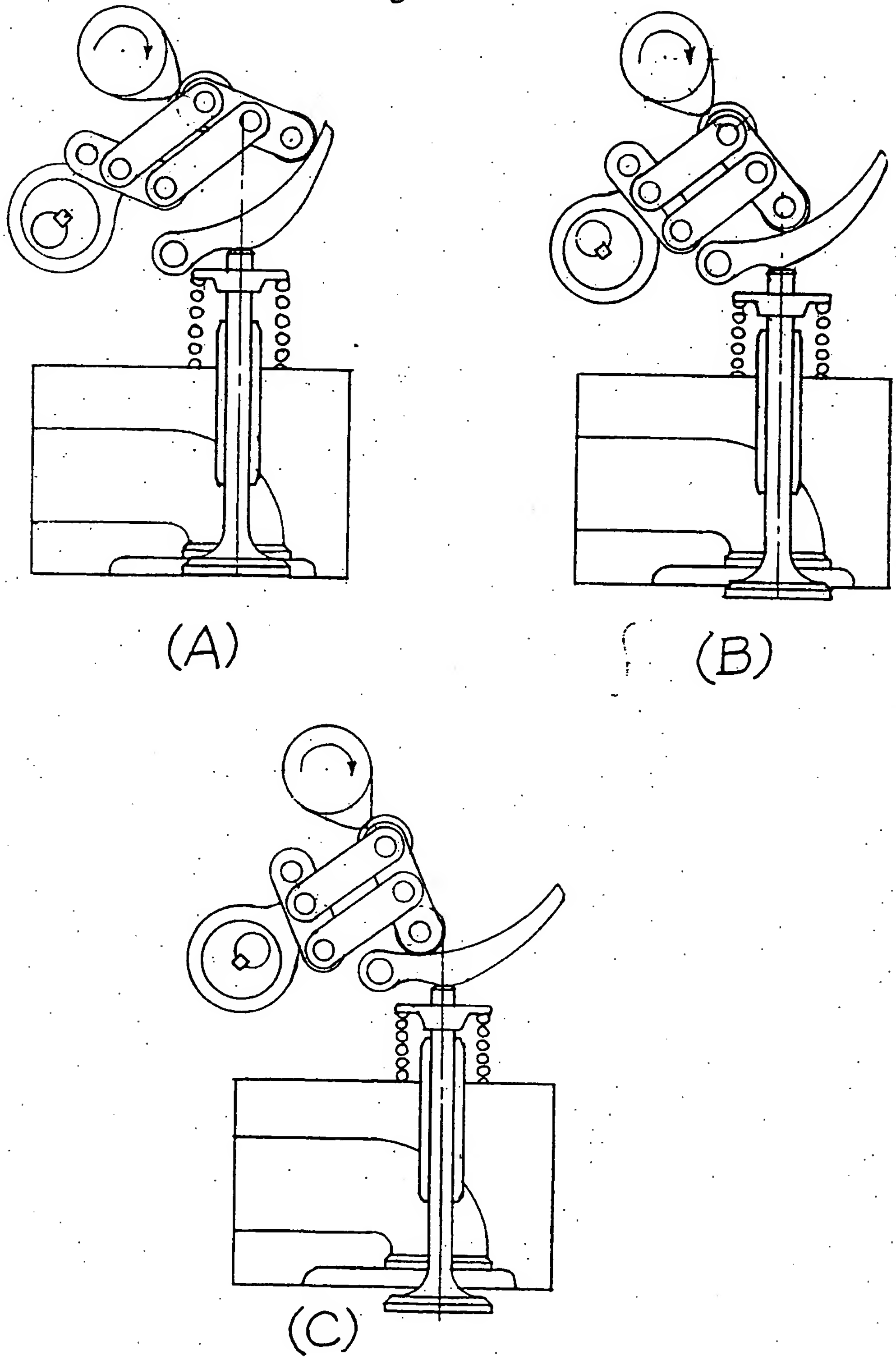


fig. 15

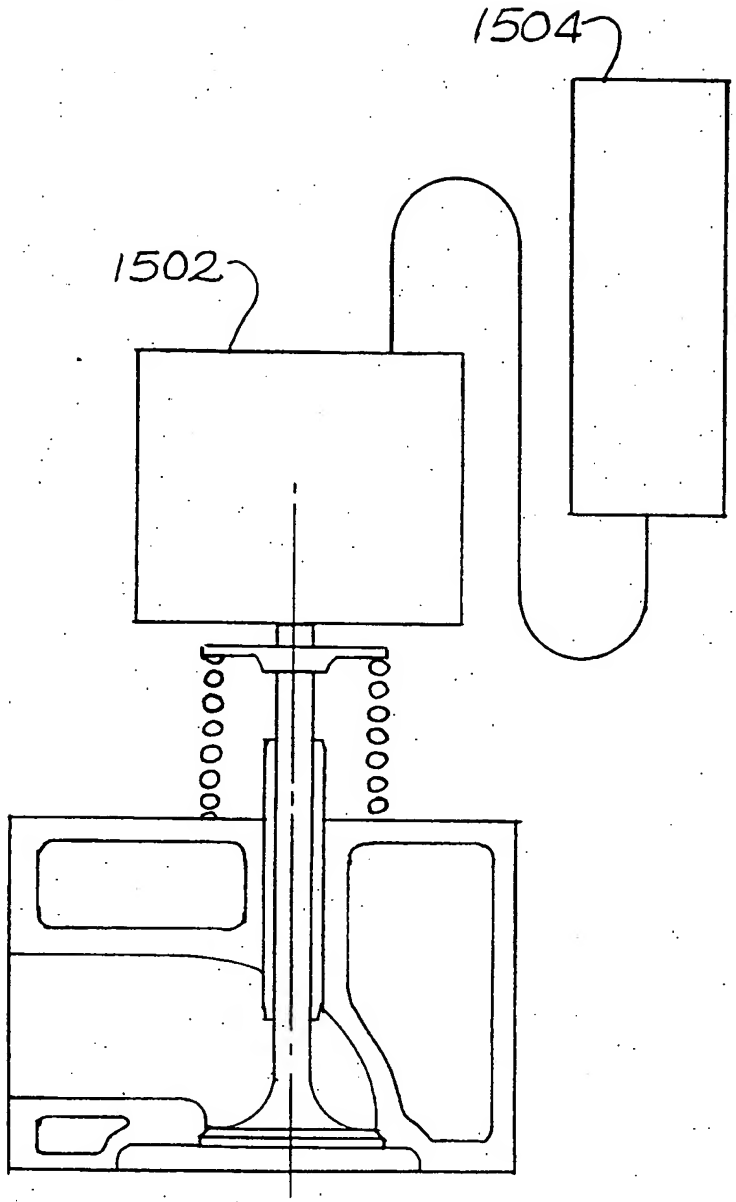
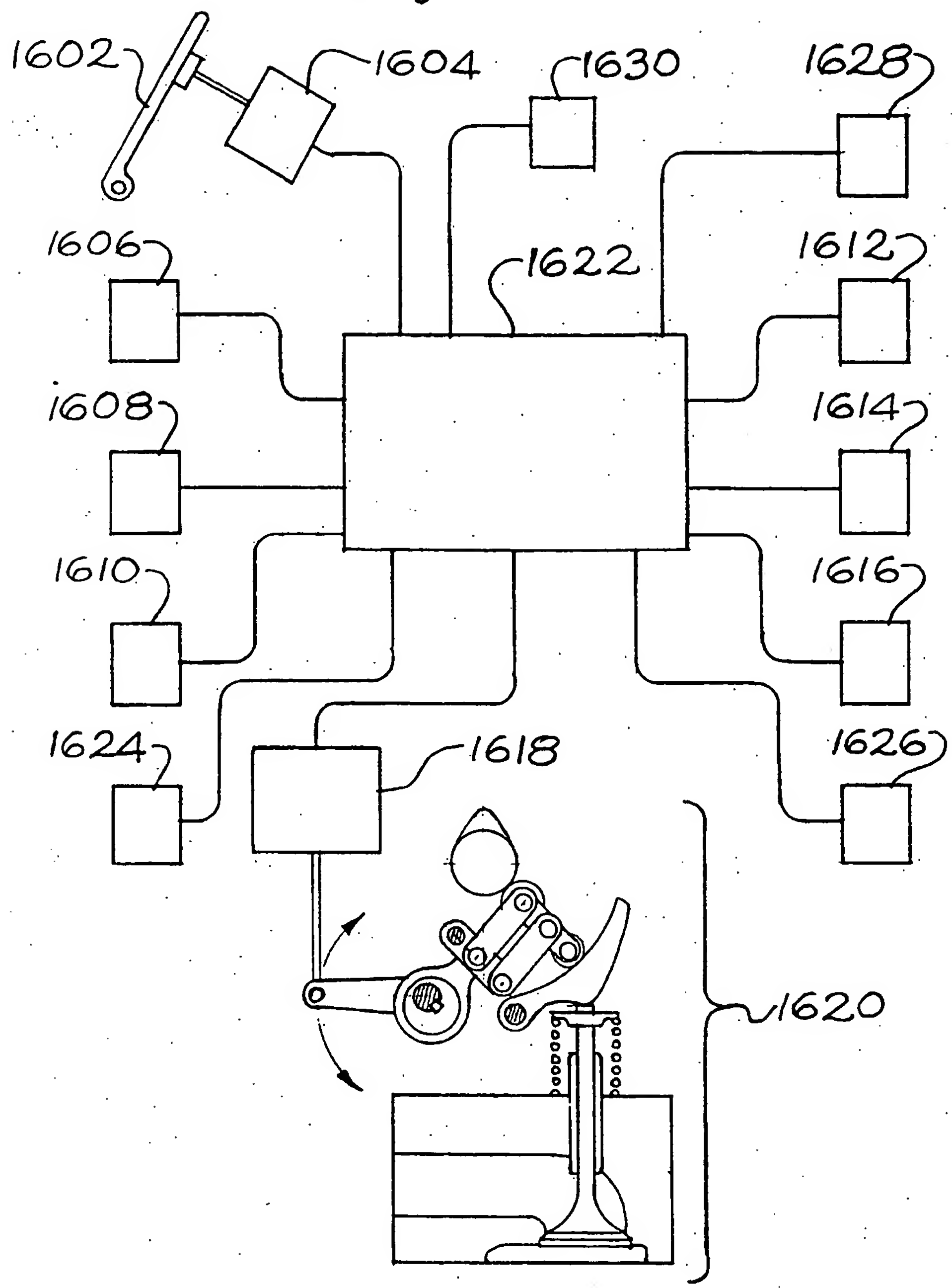


fig 16


1602
1604
1630
1628
1606
1622
1612
1608
1614
1610
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1624
1618
1626
1620

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US00/11933

A. CLASSIFICATION OF SUBJECT MATTER
IPC(7) :F02N 3/00; F02B 75/02; F01L 9/04, 9/02, 1/34
US CL : 123/188.2, 90.1, 316
According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
U.S. : 123/188.1, 188.3, 188.4, 188.7, 188.8, 90.11, 90.12, 90.14, 90.16

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X --- Y	US 4,106,466 A (GOLOFF) 15 August 1978, Figures 1-3.	1-3, 7, 14 --- 8-13, 23-38
X --- Y	US 1,415,374 A (LOVEJOY) 09 May 1922, Figures 1 and 2, lines 29-51 of column 1 of page 3.	1-3, 6, 7, 20-22 --- 8-13, 15-19, 23-38
X --- Y	NR 46,101 (HINSCH) 11 February 1929, Figures 1-4.	1-3 --- 5, 8-13, 23-38

[X] Further documents are listed in the continuation of Box C. [ ] See patent family annex.

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

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Date of the actual completion of the international search 08 AUGUST 2000	Date of mailing of the international search report 13 OCT 2000
Name and mailing address of the ISA/US Commissioner of Patents and Trademarks Box PCT Washington, D.C. 20231 Facsimile No. (703) 305-3230	Authorized officer THOMAS E. DENION Diane Smith for Telephone No. (703) 308-2623

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X --- Y	NR 366,423 (SCHMIDT) 15 February 1963, Figure 3.	1, 2, 4 --- 5, 8-13, 23-38
Y	US 2,988,080 A (RANKL) 13 June 1961, Figure 7.	10, 11
Y	US 5,357,914 A (HUFF) 25 October 1994, Figure 1.	15-19
Y	US 5,230,315 A (KANESAKA) 27 July 1993, Figure 1, Abstract.	23-26, 28
Y	US 5,123,388 A (KANESAKA) 23 June 1992, Figure 1, col. 2, lines 35-37.	23-26, 29
Y	US 4,009,695 A (ULE) 01 March 1977, Figures 1, 2, 5-11.	25, 26, 30-34
Y	US 5,611,303 A (IZUO) 18 March 1997, Figure 1.	27
Y	US 5,419,290 A (HURR et al.) 30 May 1995, Figures 1, 1A, 2, 2A, 11.	35-38
Y	US 5,373,818 A (UNGER) 20 December 1994, Figures 5-7.	35-38